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DEVELOPMENT OF NOISE REDUCTION KITS  
FOR THE U.S. ARMY  
10,000 LB ROUGH TERRAIN FORKLIFT TRUCK

Final Technical Report

By: Robert N. Baker

June 1974

U. S. ARMY MOBILITY EQUIPMENT RESEARCH  
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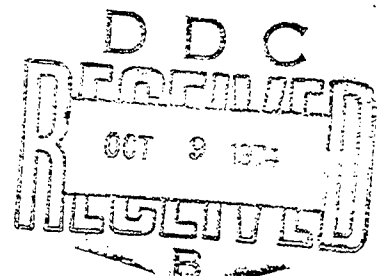
H. L. Blachford, Inc.

Troy, Michigan 48084

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### ABSTRACT

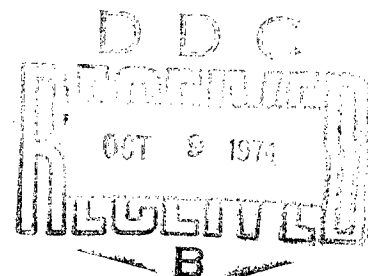
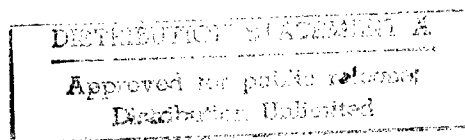
Noise at the operator's ear of the Rough Terrain Forklift Truck (RTFT) was found to be excessive. A program was initiated with goals to develop retrofit kits to reduce sound level from over 100 dB(A) to 90 dB(A) without a cab or degradation of performance and also to 85 dB(A).

Noise sources were identified and quantized by using sound and vibration measurements, spectrum analysis and selective component operation. Combustion air intake noise, exhaust noise, airborne noise from openings in the engine compartment and vibration of panels near the operator due to hydraulics, engine and transmission vibration were most important.

Changes were made to reduce noise of each important contributor. The air cleaner was moved away from the operator, damping material was added and an extra intake noise silencing element was installed. Two very large exhaust mufflers replaced the originals on the engine hood behind the driver; double shells limited radiated noise. Noise due to panel vibration was treated by damping, isolation mounting of the engine and transmission, use of flexible hydraulic suction lines, replacement of one hydraulic pump and isolation mounting of the hydraulic reservoir to minimize interaction with other panels. The cooling fan was replaced by one with larger blades to provide equivalent cooling at reduced speed. Numerous special enclosure parts were installed and sound absorption pads were added to the inside of the RTFT.

These changes reduced noise to about 90 dB(A). The 85 dB(A) package required adding a noise shield behind and to the right of the operator.

Cost to reduce noise to 90 dB(A) was estimated at \$500 for parts and 85 hours labor. The 85 dB(A) package estimate was \$720 and 90 hours.



## SUMMARY OF RESULTS

The body of this report documents not only the various results achieved in this project, but also the entire investigative sequence in roughly chronological order. About 80 graphs and tables are discussed. The purpose of this summary is to concisely describe only the final, successful treatment of each noise source or path area and the noise reduction achieved. It is hoped that this will provide a quick overview of the entire program and a compact introduction to the areas of investigation of most interest.

The noise at the operator's ear of the Rough Terrain Forklift Truck (RTFT) was found to be excessive relative to the new MIL-STD 1474, "Noise Limits for Army Materiel". A program was initiated with a goal of developing retrofit kits to reduce the sound level from over 100 dB(A) to 90 dB(A) without the use of a cab or degradation of performance and also to 85 dB(A).

### Source Identification

Noise sources in the vehicle were identified and quantized. The so called "window" technique was employed primarily. Narrow band frequency peaks were related to the known operating (e.g. rotational) frequencies of the various components. Numerous vibration measurements were also made and the spectra of these were compared to the noise spectra and operating frequencies.

Sound at the operator's ear position of the RTFT was primarily controlled by combustion air intake noise, exhaust noise, airborne noise from numerous openings between the operator station and the engine/transmission compartment, and vibration of panels near the operator. The panels are caused to vibrate due to excitation from the hydraulic system, engine and transmission. Engine cooling fan noise and resonances of some thinner sheet metal panels are lesser, but still important contributors to operator ear noise.

Modification of the vehicle with only available "off-the-shelf" quiet components was ineffective because uncontrolled sources and paths such as holes near the operator established the minimum to which sound level could be reduced.



### Intake Noise Reduction

In the RTFT as received, the air cleaner was mounted on the left rear fender directly behind the operator. The "tooth" frequency of the Roots blower produced a prominent annoying peak in the intake noise spectrum. A less serious, but still important problem was shell radiation due to vibration of the air cleaner. The solution chosen was to add damping material to the air cleaner, to move the intake to the opposite fender and to install an extra silencing element in the intake system between engine and air cleaner.

### Exhaust Noise Reduction

Most of the exhaust noise was due to shell radiation from the twin mufflers. These are mounted on the hood of this rear engine vehicle behind the operator. Each of several "wrapped" mufflers evaluated limited the radiated noise adequately but the remaining airborne noise problem was more difficult to solve. Two "off-the-shelf" replacement units produced no improvement. The final choice was a pair of mufflers of conventional design but greatly increased volume and double wrapped shell specially designed to use all available space on the hood.

### Panel Vibration Reduction

Noise due to the vibration of the panels near the operator was excessive. Some panels were successfully treated by damping. All vibration levels were reduced by isolation mounting of the engine and transmission.

The surfaces closest to the operator were also excited to vibration by the several hydraulic gear pumps in the vehicle for power steering, power brakes and the lift system. Both structure-borne energy through the rather rigid suction lines and fluid-borne energy through the return lines were important. The greatest source of fluid-borne excitation was found to be the engine mounted power steering pump.

The final redesign involved flexible suction lines, replacement of the power steering pump and isolation mounting of the hydraulic reservoir to minimize interaction with other panels.

### Engine Cooling Fan Noise Reduction

Two alternative cooling fans were tested. One was a 6 blade, 24" diameter "pusher" fan with integral ring shroud. The "U" faced outward in the fashion of a bicycle wheel. It was overlapped radially by a fixed orifice shroud by 1/2". The result was zero blade tip clearance. Another alternative was a 6 blade, 26" diameter fan drawing air into the engine compartment through the radiator. This fan had larger blades and modified blade pitch to achieve a more uniform velocity profile over the fan blade. Cooling with the ring fan was equivalent to the original fan with no speed increase while the large blade fan provided equivalent cooling at 15% less rotational speed. For this application the second fan produced the greatest noise reduction and it was selected.

### Enclosure and Acoustical Materials Treatment

None of the above discussed source treatments would have been worthwhile without modification of the noise paths. Numerous sheet metal and sound barrier material components were added to improve the enclosure and sound absorption was added to the inside surfaces of the vehicle. Finally, the 85 dB(A) package included a partial shield behind and to the right of the operator. It achieved a further reduction of 4-6 dB at the operator's ear.

### Summarized Results

The results of this work are presented in greatly simplified form in the following table of lab test data. It divides the noise at the operator's ear into 6 source areas showing their strength before and after installation of the 90 dB(A) package under High Idle conditions. Also shown is the 85 dB(A) package final High Idle level.

<u>Condition</u>	<u>Total</u>	<u>Source</u>					
		<u>Exhaust</u>	<u>Intake</u>	<u>Panel</u> <u>Vibration</u>	<u>Fan</u>	<u>Nearby</u> <u>Open</u> <u>Holes</u>	<u>Other</u> <u>Airborne</u>
As Received	101 dB(A)	97	95.5	91.5	84	97	89
90 dB(A) Kit	90 dB(A)	78	78	85	78	80.5	86
85 dB(A) Kit	86 dB(A)	--	--	--	--	--	--

Budgetary quotes from vendors and estimates made by H. L. Blachford personnel produced the following estimates of parts cost and installation and rework labor:

	<u>Parts</u>	<u>Hours Labor</u>
90 dB(A)	\$500	85
85 dB(A)	\$720	90

#### Conclusions

This program has demonstrated that significant (10-15 dB(A)) noise reduction of this type of vehicle can be achieved using conventional analysis and re-design procedures and techniques without degradation of performance and at reasonable cost.

In future programs to develop more than one type of noise control kit, we suggest that the kit of greatest noise reduction be developed first and lesser kits be designed using the most cost-effective parts of the major kit. This will lead to shorter development schedules and make the task of minimizing kit cost easier.

## FOREWORD

This work was conducted under Award/Contract Number DAAK02-73-C-0473 from the U. S. Army Mobility Equipment Research and Development Center, Fort Belvoir, Virginia 22060.

Valuable support and assistance was provided throughout by Mr. Samuel Wehr, MERDC Technical Representative, by Detroit Diesel Allison Division of General Motors Corporation and by the engineering personnel of the Dana Corporation Technical Center.

Many H. L. Blachford personnel worked diligently on this program including Miss Sherrill Sargent and Messrs. Franklin Barnes, James Bliss, James Groening, James Masiak, David Patterson and Thomas Saur.

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## INTRODUCTION

The U. S. Army's 10,000 lb. Rough Terrain Forklift Truck (RTFT) is a large load handling and towing vehicle weighing over 34,000 pounds. It has selectable two or four-wheel drive and steering. Power is provided by a rear mounted two cycle V-6 Diesel Engine with a Roots type blower. A short propeller shaft connects the engine to the three speed automatic transmission. The vehicle is capable of operating in up to 5 feet of surf. The vehicle has no operator cab or really effective noise barriers between the operator and the various noise sources in the machine.

A comprehensive noise reduction program was undertaken with two goals. One was to develop a retrofit kit to reduce operator location sound level from over 100 dB(A) to 90 dB(A) without the use of a cab or degradation of performance. A second goal was to further reduce noise to 85 dB(A) using a cab if necessary. Any performance modification to achieve 85 dB(A) was to be fully coordinated with the U.S. Army Mobility Equipment Research and Development Center.

The technical effort was divided into five phases as follows:

### INVESTIGATION PHASES

- Phase I: Baseline Field and Laboratory Tests
- Phase II: Characterization of all Significant Sound Sources
- Phase III: Development of 90 dB(A) Kit Components
- Phase IV: Final 90 and 85 dB(A) Kit Development
- Phase V: Final Field Test Data

As the program actually evolved, there was some overlapping of Phases II and III and considerable overlapping of Phases III and IV.

Following this introduction the major sections of this report are Investigation, Discussion, Conclusions and Recommendations.

## INVESTIGATION

### Phase I: Baseline Field and Laboratory Tests

#### Objectives:

One objective of Phase I was to conduct tests to measure the sound at operator and spectator locations in a variety of operating conditions. Another objective was to document the cooling system performance and drawbar pull force capability under stabilized conditions. These data were required as a reference against which to compare cooling and drawbar pull after changes had been made to reduce noise at the operator station. The final objective was to repeat the operator station measurements in our laboratory so that differences between lab and field data could be corrected and to make any other baseline measurements which seemed appropriate.

#### Field Test Location:

Measurements were made with the vehicle out-of-doors at the Dana Corporation Technical Center test track near Ottawa Lake, Michigan. Performance tests of drawbar pull and engine cooling were made in accordance with SAE Standard J872a, "Reserve Tractive Ability" and SAE Recommended Practice J819a, "Engine Cooling System Field Test Code". These tests were made on the 1.75 mile 3 lane concrete oval test track. Dana corporation personnel conducted these tests using their instrumentation which is described in Appendix C.

Baseline sound measurements were made on the asphalt turn-around pad within the oval track, and at the entrance to it, with the vehicle moving from West to East on the inside lane of the track. Figure 1.1.1 shows plan views of the Dana Corporation Technical Center and the specific area of sound measurement work.

All sound measurements were made by H. L. Blachford personnel. Weather was clear and dry with winds never exceeding gusts of 12 miles per hour. Measurements at the operator's ear position were made in accordance with SAE Recommended Practice J919a, "Sound Level Measurements at the Operator Station for Agricultural and Construction Equipment". Instrumentation included a Bruel and Kjaer Type 2204 Impulse Sound Level Meter with Type 4134 1/2 inch Pressure microphone and windscreen. The signal from the sound level meter was fed to a Nagra Kudelski Tape Recorder, Type IIIN for future analysis. The pressure microphone was oriented upward in this and all operator ear tests for the most uniform frequency response. It was supported by a modified shoulder holster worn by

the operator to keep it a constant distance from his head and to minimize the vibrational inputs from the vehicle.

Spectator noise measurements were made in accordance with SAE Standard J952b, "Sound Levels for Engine Powered Equipment". Instrumentation included a Bruel and Kjaer Type 2206 Sound Level Meter with Type 4148 1/2 inch free field microphone and windscreen. The signal from the sound level meter was fed to the Nagra Kudelski tape recorder for future analysis.

Calibrations were made before and after measurements at each position using a Bruel and Kjaer Type 4220 Pistonphone calibrator. Background sound level was below 50 dB(A) during all tests and thus more than 20 dB below vehicle measurement levels for all positions and test conditions. The instrumentation systems used met the requirements of SAE Recommended Practice J184, "Qualification of a Sound Data Acquisition System".

#### Laboratory Test Location:

The RTFT was tested in the Blachford Acoustical Laboratory Semi-anechoic Room, a large anechoic half-space, measuring approximately 25 feet by 45 feet by 16 feet high clear space. The floor is concrete and the walls and ceiling are treated using fiberglass wedges of 2 foot depth. This environment is well suited for development work on vehicles since the room simulates an out-of-doors environment with a hard reflecting ground plane with anechoic half-space above it. The room is equipped with a silent air change system with a make-up air heater, capable of flowing 18,000 cfm through the room to purge it of exhaust and other fumes and to prevent excessive variations in room air temperature. Figure 1.1.2 describes the laboratory test location.

Sound measurements in the laboratory were made using a Bruel and Kjaer Type 4134 1/2" pressure microphone cabled to a Bruel and Kjaer Type 2603 Microphone Amplifier. The signal from the amplifier was fed to either an Ampex Tape Recorder, Type 602-2 or an Ampex Tape Recorder Type AG440 for a permanent record and future analysis. These systems also met the requirements of SAE J184.

#### Baseline Test Data and Discussion:

Detailed results of the performance tests of drawbar pull and engine cooling are presented in Figure 1.18. This data is discussed in the Phase V section of this report where it is compared to the final performance data.

Six operating conditions were used for noise tests as follows:

- 1) Idle - stationary vehicle, transmission in neutral, engine speed approximately 500 revolutions per minute (RPM).
- 2) High Idle (HI) - stationary vehicle, transmission in neutral, accelerator pedal fully depressed, engine speed approximately 2950 RPM.
- 3) Torque Converter Stall (TCS) - stationary vehicle, transmission in high gear, accelerator and brake pedals fully depressed, engine speed approximately 2800 RPM.
- 4) High Idle Raising Maximum Rated Load (10,000 lbs) or High Idle Raising 8000 Lb. Load - Stationary vehicle, transmission in neutral, lifting load with forks, engine speed approximately 2950 RPM.
- 5) Intermediate Gear Loaded to Rated Speed (IGIRS) - Vehicle travels at uniform speed in 2nd gear, accelerator fully depressed, brakes applied to slow engine speed to 2700 RPM, no load on forks.
- 6) Full Throttle Acceleration Per SAE J88, "Exterior Sound Level Procedure for Powered Mobile Construction Equipment" - unloaded vehicle initially traveling in high gear at 2100 RPM engine speed. Throttle is fully opened as rapidly as possible and vehicle accelerates for a distance of 66 feet.

Before discussing the baseline noise data some comments on the numbering system used for the figures in this report is appropriate. Sketches, tables and graphs relating to noise data are in a decimalized numerical sequence. The first numeral denotes the phase of the program, 1 through 5. The second numeral identifies the figure within that phase. If two figures are especially closely related then they have both first and second numeral common with a third numeral to separate them. See Figures 1.1.1 to 5.4.

Figures showing vibration measurement data follow the noise data. They are in numerical sequence with the prefix "V". See Figures V1 to V13. Detailed results of the field noise tests are presented in Figures 1.2 through 1.10. Figures 1.11 through 1.17 present laboratory results. The following table summarizes the A-weighted levels at the operator's ear. Octave band data is presented in Figures 1.2, 1.7 and 1.11.

<u>Condition</u>	<u>A-Weighted Sound Level</u>	
	Field Test	Lab Test
Idle	82.5 dB(A)	81.5 dB(A)
High Idle (HI)	102.5	100.5
Torque Converter Stall	102.5-106	102.5
HI, Raising Max. Rated Load	100.5	-
Intermediate Gear Loaded to Rated Speed	105.5	-
Full Throttle Acceleration Per SAE J88	106.5	-
HI, Raising 8000 Lb. Load	-	100.5

Spectator noise 50 feet from the stationary vehicle was tested in accordance with SAE J952b and found to range from 62 to 64 dB(A) at idle and from 82 to 87 dB(A) for the three conditions of High Idle, Torque Converter Stall and High Idle Lifting Maximum Load (Figure 1.3 to 1.6).

Pass-by noise tests were made in accordance with SAE J952b under conditions of Intermediate Gear Loaded to Rated Speed and Maximum Vehicle Speed. These levels ranged from 85 to 91.5 dB(A) (Figures 1.8 and 1.9).

"ISO-SPL" lines<sup>1</sup> were determined by laying out a set of 8 axes with origin at operator's ear. The results of this test are shown in Figure 1.10. The microphone was at the ear level of a standing observer.

Using the same measurement system used at the operator's ear position and the same upward microphone orientation, additional tests were made in the Semi-Anechoic Room at H. L. Blachford, Inc. and the data are shown on figures 1.11 through 1.17. Measurement positions were as follows:

4 feet from vehicle	1.12
Engine compartment (near steering pump)	1.13
In body under operator (near transmission)	1.14
Near engine exhaust	1.15
Near engine fan	1.16
Near air intake	1.17

---

1. "ISO-SPL" lines are defined by the U.S. Army MERDC as the equal sound level lines in the horizontal plane plotted in 5 dB(A) steps from the High Idle sound level at the operator's station to 15 dB(A) less than that level.

The levels measured at the operator's ear were higher than had been anticipated, the greatest being about 106 dB(A). This meant the final kits would have to achieve very substantial reductions to meet the goals of 90 and 85 dB(A).

In the field tests, the two noisiest conditions involved a moving vehicle and these tests could not be conducted during the lab development work. In spite of this limitation it appeared that the stationary tests produced levels close enough to those of the moving tests so as not to be misleading in the source identification and redesign work.

#### Phase II: Characterization of all Significant Sound Sources

##### Objectives:

One objective of this phase was to identify all sources of sound which contribute significantly to sound at the operator's ear location. A second objective was to develop an understanding of the paths by which energy from each significant source reached the operators ear location. A third objective was to determine the improvement possible by substitution of available off-the-shelf components only.

##### Window Technique:

The experimental approach used primarily in Phase II has been termed the "window" technique. In this technique, experimental noise controls are applied to all known sources of noise, and one control at a time is removed (window) and that source measured. This approach yields success if properly used by experienced noise control engineers. It is not a technique which can be used to evaluate all noise sources, but it is often a good way to sort out sources and the many parallel paths of sound and vibration.

One problem with the window technique is that the investigator initially does not know what all the sources will be nor does he know their relative importance. This introduces an element of trial-and-error into the procedure. The approach in this program was to develop a sequence of sets of controls of "treatment packages". The initial package was designed to control the most common strong vehicle noise sources of intake, exhaust and engine cooling fan. Subsequent treatment packages included every greater degrees of source elimination or control, enclosure, vibration isolation, sound absorption and vibration damping.

Another problem is that the control introduced may be inadequate if the source strength is underestimated. (This happened in the case of air intake noise.) Using other techniques such as spectrum analysis and vibration measurements helps to point up control deficiencies.

Selection of the "windows" to be opened from the fully treated condition may be somewhat arbitrary. Sources or paths may be grouped to try to ensure that opening the window will have a measureable effect.

#### Spectrum Analysis Technique:

All laboratory data was tape recorded. Almost all of it is presented as octave band data in this report, but many conditions were also analyzed using a 6% constant bandwidth narrow-band analyzer. Such analysis allows the investigator to compare prominent peaks in the frequency spectrum with the known operating frequencies of various machine components.

This was done for the RTFT. The results of computing operating frequencies are listed below for High Idle at an engine speed of 2930 RPM. Only fundamental (1st harmonic) frequencies are listed.

<u>Event</u>	<u>Frequency, Hz</u>
Engine rotational	49
Engine firing	293
Engine fan blade pass	244
Roots blower pulsation	484
Hydraulic (9 tooth) gear pump pulsation	440
Torque converter pump blade pass	976
Torque converter stator blade pass	1074(approximate)
Torque converter turbine blade pass	1122(approximate)

#### First Treatment Package:

Figure 2.23 defines the composition of the various treatment packages used in the source identification tests. Figure 2.24 is a sketch showing treatment locations. Frequent reference to these figures should help in understanding the data of the other figures relating to Phase II.



Initially the vehicle was treated by running remote air intake and exhaust lines and turning the engine fan off. The muffler shells were "wrapped" with a fiberglass mat and a limp barrier material<sup>2</sup> to reduce radiated noise. The barrier material was also used to wrap all remote intake and exhaust lines. These steps eliminated three major air-borne noise sources. The result was only about 2 dB(A) reduction from the as-received levels of 100.5 to 102.5 (Figure 1.11). The reason was the large number of open holes in the lift control panel, operator floor, and upper boom area allowing direct passage of sound to the operator's ear from the engine compartment. A barrier treatment was devised to close these holes. It, together with the intake, exhaust and fan noise controls, constituted the "First Treatment Package". This reduced sound levels at the operator's ear from 100.5 to 102.5 (Figure 1.11) to 91.5 to 92 dB(A) (Figure 2.1).

Elements of this package were removed one at a time. First the vehicle was returned to the production exhaust conditions (Figure 2.2). Next with remote exhaust lines restored the barrier treatment was removed. Refer to Figure 2.3. Leaving the barrier treatment off so as to not "unfairly" block any noise paths from the fan through the engine compartment the fan was turned on (Figure 2.4). Fan noise did not change the levels in this condition. It was overpowered by the noise of the engine and transmission.

#### Second Treatment Package:

The next step was installation of the "Second Treatment Package" to eliminate all airborne noise paths from within the vehicle. This included those measures mentioned above as well as sealing all engine compartment openings with close fitting 5/8 inch thick plywood plates and dense formable clay for sealing duct work and barrier material. Sound absorption was added to the engine compartment in the form of 1" and 1.5" thick urethane foam covered with .004" thick vinyl. Engine side panels were sealed to the body; see Figure 2.5. There was no noticeable noise reduction! This was in spite of the fact that the sound level in the engine compartment was reduced 5 dB by the absorption treatment (Figure 2.6). The data of figure 2.5 indicated that the operator ear noise level was being controlled by vibration of the sheet metal panels since elimination of the airborne noise paths had no effect. Both engine and transmission in the RTFT were mounted directly to the vehicle frame. Removal of the propeller shaft disabled the torque converter, hydraulic pump drive gears,

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2 A composite barrier material consisting of 1/4" urethane foam "decoupling" layer bonded to a 0.8 lb/ft<sup>2</sup> impervious barrier layer was used for barrier enhancement throughout the program when required. A heavy furnace duct tape was used to install the material. Double layers of this treatment were generally used on surfaces near the operator.

and two of the three hydraulic pumps, the main lift system pump and the lock steering/hydraulic brake pump. This produced a reduction in High Idle noise level of 4.5 dB(A) (Figure 2.7). Isolation mounting of the engine reduced operator ear noise by 3.5 dB to 85 dB(A) (Figure 2.8).

#### Third Treatment Package:

It was desired to drive the "residual" noise level as low as possible without disabling the transmission and hydraulic system. This would allow us to make tests to determine the degree of enclosure required. This led to the "Third Treatment Package". This package added coverings of limp barrier material to all sheet metal panels near the operator as well as to the air intake canister which vibration measurements indicated might be radiating noise quite strongly from its position immediately behind the operator. This package also included isolation mounting of the engine and transmission on neoprene pads fabricated from machinery mounting pad material. Pad sizes were designed such that they would be loaded to the manufacturers recommended load at 50 p.s.i. or 100 p.s.i. The particular materials used were Fabcel 50 and Fabcel 100. These are 5/16 inch thick neoprene sheet with molded-in pockets. Static deflection in compression (vertical) was designed to be between 0.05 and 0.06 inch. From this condition controls were selectively removed to assess the contributions of transmission vibration, air intake, radiator opening, fan (with enclosed engine compartment) and the open underside of the vehicle (Figures 2.9 to 2.13). It was discovered by use of narrow band spectrum analysis that the 500 Hz octave band was still being controlled by engine blower noise, even with the remote, wrapped intake line. Further vibration measurements indicated that the effect of the hydraulic pumps with their tooth frequency of about 440 Hz at high idle was possibly being obscured by this problem.

#### Fourth Treatment Package:

The remote intake line was redesigned moving the Donaldson air cleaner away from the operator to the floor of the semi-anechoic room. Intake lines near the operator were wrapped more heavily. This was the "Fourth Treatment Package" (Refer to Figures 2.23, 2.24)

In this condition the effects of vibration and leakage from the panels near the operator (not including the lift control lever panel or hood) were studied. This is shown in Figure 2.14<sup>3</sup>. First the areas below and then also above the boom were opened (Figure 2.15). The airborne noise from the underside of the vehicle was again examined, (Figure 2.16). Finally, the seals were removed from the engine side panels (Figure 2.17).

#### Fifth Treatment Package:

Narrow band analysis again showed that air intake noise around 500 and 1000 Hz was still a problem. The entire intake system was moved to the right side of the vehicle thus placing it much further from the operator compared to the production location. The lower boom area and underside of the vehicle were opened since closing them had little effect. This allowed a study of hydraulic system noise in a more realistic condition than fully sealed. This was the "Fifth Treatment Package". (Refer to Figure 2.23, 2.24)

Figure 2.18 shows the importance of noise at the operator's ear due to the hydraulic system and transmission. The bottom curve (3) shows the sound levels with the prop shaft off, the three hydraulic pumps removed and the hydraulic lines drained and dis-connected. Essentially only engine noise remains. Connecting the transmission raises the level 2.5 dB(A). This is shown by curve (2). The hydraulic system airborne noise raises levels another 2.5 dB(A). This is curve (1) or the normal fifth treatment package. Finally curve (4) taken from Figure 2.14 demonstrates the great importance of panel vibration due to the hydraulic system as shown by uncovering the panels near the operator.

Figures 2.19 and 2.20 show noise levels near the hydraulic pumps. Figure 2.19 give data for the same position as Figures 1.13 and 2.6 and is 8.5" from the engine mounted hydraulic power steering pump. Figure 2.20 shows noise levels 6" below the transmission mounted gear box which drives the main lift system and hydraulic brake/lock steering pumps. This position was necessarily quite near the hard, reflective floor of the room.

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3 Note that there is considerable fluctuation in the 63 and 125 Hz octave bands. The reasons for this are not known but these bands have negligible effect on the A-weighted sound levels and can virtually be ignored.

Figures 2.14, 2.18, 2.19 and 2.20 establish that the hydraulic system is an important part of the noise problem. They do not show which is most important among the hydraulic steering, braking, and fork lift systems. This question was answered by examining the vibration data.

#### Vibration Data and Discussion:

When sound levels are known to be largely a result of the vibration of machine surfaces, it is convenient and efficient to use measurement of vibration of those surfaces to evaluate the utility of modifications or treatments. The results of anti-vibration treatments such as isolation and panel damping in particular should be first studied by comparing vibration levels. Also, due to the presence of airborne sound paths which obscure the data, sound measurements may not show the effect of useful anti-vibration measures. As an example of convenience, the effect on vibration of introducing an isolator can be measured whether or not other sources such as (say) the exhaust have been adequately silenced. It is also possible to evaluate the amount of sound near a panel due to vibration of the panel; one assumes the acoustic particle velocity is the same as panel velocity and sound pressures can be calculated.

Figure V.1 shows acceleration levels on the vehicle frame near an engine mount, Engine isolation had a pronounced effect at low frequency but at higher frequencies excitation from the hard mounted transmission still predominated. Isolation of the transmission led to a further reduction in acceleration levels at all frequencies. Figure V.2 shows the resulting change in vibration of the right front fender. This point was considered relatively free of excitation due to mechanical vibration "shorts" from hydraulic lines, control linkages or exhaust lines and therefore controlled primarily by excitation through engine and transmission mounting points. Isolation produced a respectable 14 dB reduction in acceleration level in the fender.

Figures V3.1 and V3.2 show acceleration levels on the top of the hydraulic reservoir immediately behind the operator. Several points stand out:

- (1) Engine and transmission isolation had little effect.
- (2) Removal of the propeller shaft which stops the torque converter pump, transmission oil pump, hydraulic pump drive gears and hydraulic lift system pumps also had small effect; levels dropped 3-4 dB.

- (3) Removal of the entire hydraulic system including the power steering system dropped the A-weighted acceleration levels nearly 20 dB.

Figures V4.1 and V4.2 show similar results as well as the effect of a panel damping treatment on the vertical panel near the operator's right leg. This panel was rigidly bolted to the reservoir. For both these measurement points, the problem was excitation of the reservoir from the hydraulic lines attached to it, and roughly half of this was due to the power steering system.

Figures V5 and V6 show different results for two other large panels near the operator, the engine hood and lift control lever panels. In both these cases panel damping produced substantial reductions in acceleration levels. Eliminating the hydraulic lift and brake systems had essentially no effect. Further, elimination of the hydraulic steering system reduced levels only 2-3 dB.

To summarize:

- (1) Engine and transmission isolation was effective in reducing vibration in the frame and wheel areas not "shorted" by hydraulic lines.
- (2) Engine and transmission isolation was not very effective in reducing vibration in panels near the operator because they were excited by the hydraulic system.
- (3) Vibration of the stiff panels nearest the operator was controlled primarily by excitation from the hydraulic system, especially power steering.
- (4) The thinner hood and lift control lever panels evidenced problems of resonance which could be controlled by damping.

Calculation of Source Strengths:

Refer to the noise data of Figures 2.1 to 2.20. A technique of logarithmic subtraction<sup>4</sup> of A-weighted sound levels can be used to quantize the importance of various sources and paths of noise to the operator ear. The results are tabulated below. All are based on High Idle operation.

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<sup>4</sup> Refer to Appendix D for an explanation of this technique and a sample problem.

<u>Source/Path</u>	<u>Calculated Contribution dB(A)</u>
(1) Engine exhaust	97 dB(A)
(2) Air intake	95.5
(3) Engine, transmission and hydraulic system airborne noise through open holes near operator	97
(4) Panel radiation due to excitation by hydraulic system	89
(5) Panel radiation due to excitation through engine mounting points	85
(6) Panel radiation due to excitation through transmission mounting points	85
(The following airborne noise components were assessed with the holes near the operator sealed, absorption in the upper engine compartment and the underside of the vehicle open.)	
(7) Airborne noise due to engine fan	84
(8) Airborne noise due to hydraulic system	83
(9) Airborne noise due to engine and accessories	80
(10) Airborne noise due to torque converter and pump drive gears	79

As noted, all the above are based on High Idle operation. This seemed reasonable since, in any partially treated condition, there was little difference among noise levels at High Idle, Torque Converter Stall and High Idle Lifting 8000 Pound Load.

#### Quiet Component Testing for an "Off-the Shelf" Kit:

Simple replacement of components such as mufflers, air intake silencer or fan did not significantly change operator ear noise levels of the RTFLT because of the importance of open holes in and vibration of the panels near the operator. To control these sources required special enclosures and material treatments and at least semi-custom isolators. In addition, the air intake had to be re-located further from the operator.

Since no purely "off-the-shelf" kit could be produced, the objective to identify an off-the-shelf component noise reduction design was modified to allow the following:

- (1) Relocate the air intake to the right rear fender.
- (2) Simple materials treatments or flat sheet metal parts considered in controlling noise from open holes and panel resonance very near the operator.
- (3) Engine and transmission on isolation pads which are custom parts, but are made from off-the-shelf material.

Under these conditions the operator ear noise level was controlled by panel radiation and airborne noise from the remaining nearby openings. The levels when such a kit was simulated are shown in Figure 2.21 and 2.22. In Figures 2.21 there is a reduction of about 7 dB from the as-received condition. This treatment would require a number of special sheet metal parts.

Three other treatment variations were simulated in Figure 2.22. Only High Idle condition is shown. Each of these cases involved only acoustical materials to control noise from the largest holes near the operator and simulated a quiet fan by turning the fan off. This is curve (1) of Figure 2.22. High Idle levels were up 1.5 dB from Figure 2.21. Curve (2) of Figure 2.22 shows the result when simple flat barriers are added to the upper boom. A small improvement was made. Finally, curve (3) shows the effect of restoring the absorption treatment to the engine compartment. The High Idle level drops to 93 dB(A).

None of the kits of Figures 2.21 and 2.22 was judged cost-effective. The added effort and expense to get a 90 dB(A) kit appeared to be relatively small and levels around 95 dB(A) did not seem adequate improvement for the effort involved.

### Phase III: Development of 90 dB(A) Kit Components

#### Objectives:

The objective of this phase was to select or design practical components and treatments to reduce operator station noise to 90 dB(A). As the program evolved it became clear that the difference between the 90 and 85 dB(A) kits would be limited essentially to a partial enclosure to be described later. For the sake of clarity and compactness this section of the report deals also with some component evaluation work done during Phase IV, Final 90 and 85 dB(A) Kit Development.

#### Air Cleaner and Silencer Tests:

Initial no-flow acoustical performance comparison tests were made on the original Donaldson air cleaner, a potential Farr replacement unit and an additive duct silencer manufactured by Universal Silencer. Later in the program three other additive silencers were evaluated.<sup>5</sup> The test procedure initially employed was this. A loudspeaker in its enclosure was sealed to the Roots blower end of the inlet piping system of the Rough Terrain Forklift Truck. The speaker and pipes were sealed in a massive treated enclosure as shown in Figure 3.1.1. The open end of the piping system protruded from the enclosure. Tests confirmed that the only significant sound path from the speaker or source was through the piping system. The air cleaners were attached to the open end of the pipe. "White" noise was fed to the speaker noise source.

In order to evaluate radiation from the air cleaner shells as well as airborne sound from the inlet an anechoic termination was built for the inlets. The purpose of this device was to eliminate the airborne sound from the inlet of the air cleaner by sealing the inlet without reflecting sound waves back toward the speaker and thus increasing the sound level in the air cleaner and pipes. This allowed measurement of radiated sound through the shell with levels inside the shell the same as though the inlet were open. Construction of the anechoic termination was straightforward. It consisted of a two foot long tube of thin sheet metal. One end was open, the other was sealed with a plywood plug. The exterior was wrapped with barrier material which was also sealed. One half the interior volume was occupied by a sound absorbing foam wedge which in cross section would appear roughly as a right triangle with dimensions equal to the tube length and diameter. The "point" of the triangle was toward the open end of the tube.

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<sup>5</sup> Refer to Appendix E for detailed information on air cleaners, silencers, fans, hydraulic pumps and mufflers evaluated.



Sound measurements made near the anechoic termination showed it to radiate negligible sound when in use.

Figure 3.1.2 compares the insertion loss<sup>6</sup> of the Donaldson and Farr air cleaners. The Farr unit was obtained in the hope that it could be packaged within the engine compartment. There was insufficient space to do this but it seemed it might be superior to the Donaldson silencer when mounted externally. Certain similar Farr units were reported to perform well in Department of Transportation (DOT) sponsored testing; it was on this basis that the Farr unit was procured. As is shown by Figure 3.1.2, however, it actually was not as good as the Donaldson unit under these test conditions.

Figure 3.1.3 shows the insertion loss of the Donaldson air cleaner with the Universal Silencer added; there is considerable improvement. An inlet silencer was also fabricated for the Farr unit but this combination was inferior to the Donaldson-Universal arrangement.

Figure 3.1.4 compares radiated noise from the two air cleaners with the anechoic termination in place. There is little difference in the two units. Both radiate strongly in the critical 500 Hz area which is near the excitation frequency of the Roots blower.

These tests indicated that there was no advantage to replacing the Donaldson air cleaner with the Farr unit and the approach should be to add a silencing element to the Donaldson unit. The Universal Silencer was too large and expensive for the application and we contacted Donaldson and Nelson Muffler Company for other devices. We were able to obtain two silencers from Nelson and one from Donaldson some time later.

An improved method of evaluating these devices was sought, and a peculiarity of our lab suggested an approach. The lab has a massive common wall between the semi-anechoic and reverberation rooms. Sound transmission loss tests of panels are made by placing a sample over an aperture in this wall and attaching a small anechoic receiving chamber to the semi-anechoic room side of the wall.

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6 Insertion loss is the difference in level between sound measured a given distance from the noise source outlet and sound measured the same distance from the outlet for noise from the device under test.

The reverberation room was used as a stable, known sound source by playing broadband noise in it as is done in transmission loss testing. The aperture was sealed with a massive plate having a hole in it to match the source-side pipe on the silencing element. See Figure 3.3.1. Accurate measurements of airborne and radiated noise from the silencing element could then be made using a microphone in the semi-anechoic room. This technique was used to evaluate the noise reduction capabilities of three air inlet silencers for the RTFT. The design of each is shown in Figure 3.3.2. The Nelson Baffle Silencer and Lined Silencer were designed for the inlet of the air cleaner. The Donaldson Silencer was designed to be inserted between the blower and the air cleaner. Airborne noise data is presented in Figures 3.3.3 to 3.3.9. Figures 3.3.3, 3.3.5 and 3.3.7 show the attenuation of simple pipes with inlet diameters and lengths equivalent to those of the silencers, and the uncorrected attenuation of the silencers. Then Figures 3.3.4, 3.3.6 and 3.3.8 show the corrected attenuation of the silencers; that is, the difference between their attenuation and that of the simple pipes.

A comparison of the three corrected silencer attenuation graphs shows the strengths and weaknesses of each. The Baffle silencer is "peaky" with attenuation peaks in the 200, 400, 800, 1600, 4000 and 8000 Hz center frequency third-octave bands and rather low "valleys" between the peaks. The lined silencer provides generally good low frequency attenuation and fair attenuation above 3150 Hz but performs poorly in the important center frequencies from 630 Hz to 3150 Hz.

The Donaldson Silencer exhibited generally good attenuation from the 400 through 6300 Hz third-octave bands. It was the best of the three in terms of attenuation of airborne noise. Figure 3.3.9 compares radiated noise from the three silencers as measured at position M3 (Figure 3.3.1). Again, the Donaldson unit was superior.

Because of these results and the fact that it could be installed most easily the Donaldson silencer was selected for use in the RTFLT noise reduction kits.

#### Cooling Fan Tests:

Two alternative fans<sup>5</sup> were compared to the production fan in the Rough Terrain Forklift Truck. All fans were mounted in the truck in their normal position and driven by an electric motor placed

in a massive acoustically treated enclosure outside the engine compartment. An opening in the enclosure was provided for a V-belt to drive the Forklift Truck fan hub at 1080 RPM. An engine side panel was replaced with a plywood cover with a small opening for the V-belt to the fan hub. Noise was measured at 5 points on a semicircle (in plan view) of 10 ft. radius from the center of the radiator, in the engine compartment and at the operator's ear. Levels did not vary greatly among the 5 measurement positions on the semicircle and were generally highest at the middle point which was on the axis of the fan (to the rear of the vehicle).

The first alternate fan tested was manufactured by Hayes Albion Corporation. It had an overall diameter of 26 inches as did the other two fans but the blade diameter was only 24 inches. A U-shaped ring shroud was attached to the blade tips. A mating orifice shroud with a 25 inch diameter opening was fit into the "U" for a labyrinth-seal effect. In spite of the smaller blade diameter, this fan provided cooling air flow equivalent to the original fan and shroud. Like the original fan it was a "pusher" fan blowing air out through the radiator. The second fan tested was a 6 blade fan with larger blades manufactured by Schwitzer Division of Wallace-Murray Corporation. It was of the "Taper-Twist" or variable blade pitch variety for more uniform velocity profile over the area of the fan. (Flow tests confirmed that this was indeed so). This fan produced equivalent cooling flow at 15% speed reduction, part of its cooling advantage apparently being that it was a conventional type fan which drew air through the radiator from outside the vehicle. An orifice shroud with 1/2" blade tip clearance was used with this fan.

Figures 3.2.1, 3.2.2 and 3.2.3 compare the octave band pressure levels from the three fans at the operator's ear, 10 feet from the fan, and in the engine compartment. Figures 3.2.4, 3.2.5 and 3.2.6 give the narrow band levels 10 feet from each fan. The large blade fan was quietest at the operator ear.

The production fan produced large peaks at the first three harmonics of blade pass frequency (Figure 3.2.4). The large blade fan suppressed the second and third harmonics but not the first (Figure 3.2.5). The ring fan suppressed all three harmonics but did not do as well at high frequencies as the large blade fan (Figures 3.2.6, 3.2.1).

As mentioned above, the large blade fan can be run at a 15% speed reduction for a theoretical reduction of another 3.6 dB assuming a 60 log (speed ratio) dependance for the sound level of a fan. An added advantage is that its lower blade pass frequency is less of a problem in terms of A-weighted sound level. For all these reasons, it was selected for the kits.

### Hydraulic System Tests:

A crucial part of the noise problem in the Rough Terrain Forklift Truck was excitation of the hydraulic reservoir and near-by panels by the hydraulic pumps and lines. Because the surfaces of these parts were so near the operator, radiated noise from them was very important. Considerable work was done to determine the effect of hydraulic system modifications on acceleration (vibration) levels on the top of the reservoir and on the panel by the operator's right leg. The effects of replacing lines, replacing pumps and isolating the reservoir were studied.

The first step was to replace long lengths of rigid pipe on the suction lines from the reservoir to the pumps with flexible hose. This had a dramatic effect in reducing the high frequency vibration of the reservoir as shown by curves 1 and 3 of Figure V7. The crucial 500 and 1000 Hz octave bands appeared to go up 3 dB, however. This was possibly due to several other modifications made to the vehicle when the flexible lines were installed. But the large high frequency vibration reduction is clear.

Curve 4 of Figure V7 shows the effect of mounting the reservoir on isolation pads. The 500 and 1000 Hz levels dropped more than 10 dB relative to curve 3.

Figure V8 shows the effects of the same modifications on the panel by the operator's right leg. Each change appeared to produce a slight reduction in level. Other "inputs" to this panel may have been as important as the reservoir, however. Figure V9, for example, shows that simply unbolting the lift control lever panel caused some large changes in octave band levels.

Other unpublished research being done for MERDC identified pump manufacturers whose products were relatively low in airborne noise. Two of these power steering pumps<sup>5</sup> were evaluated in the hope that they would also produce less vibration. Both performed well. Figure V10 shows that the Tyrone unit produced higher levels only at 1000 Hz and the Commercial Shearing pump produced higher levels only at 4000 Hz as compared to no pump. Figure V11 shows that when the propeller shaft is installed and the other two Warner-Motive pumps are running the overall levels are lowest with the Tyrone pump. Note that replacement of only the steering pump (which causes only one-third of the total hydraulic flow) reduced the vibration levels by 5 dB.

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7 At High Idle the first three harmonics would be changed from 244, 488 and 732 Hz to 210, 420 and 630 Hz. The changes in attenuation due to A-weighting would be roughly 2.5, 1.0 and 0.7 dB respectively.

Use of the Tyrone pump resulted in a 3 dB reduction in vibration levels in the panel by the operator's right leg compared to the production pump. The Commercial Shearing pump produced no change in overall level of that panel.

Even with the Tyrone steering pump, airborne noise and vibration at the gear tooth frequency of the main hydraulic pump or lift pump was still a problem, especially at "High Lift" operating condition. No Tyrone pump to meet the needs of the lift system was available. One Commercial Shearing unit was available and we tested it.

Figure V12 shows the effects of installing the Commercial Shearing lift pump on reservoir acceleration levels. As was the case in the power steering pump application, the Commercial Shearing unit greatly reduced the energy at 500 Hz but did not do particularly well as higher frequencies. In fact there was an apparent slight increase in the A-weighted vibration level in the top plate of the reservoir.

To check airborne noise from this pump, the microphone was placed at the measurement position used in Phase II 6 inches below the transmission drop box. Both the lift pump and the power brake/lock steering pump are mounted to this drop box. Figure 3.3 compares the sound levels 6 inches below the drop box with the Commercial Shearing lift pump with data from Figure 2.20.

Again there was an improvement in performance at 500 Hz but higher acoustical noise levels at frequencies above 1000 Hz. Figure 3.3 also shows that there is very little difference in the noise level near the pump under conditions of High Idle and High Lift.

These results were also confirmed in tests made with the microphone at the operator's ear position; levels at 500 Hz were reduced, but those at 2000 Hz were clearly increased. The net effect was apparently a very slight increase in the A-weighted sound level. Thus, replacement of the existing Warner-Motive lift pump with the Commercial Shearing unit would have no value other than reducing the 500 Hz peak under high lift conditions at the operator ear. Therefore the 90 and 85 dB(A) packages do not include replacement of the lift system pump.

#### Muffler Tests:

In the "as-received" RTFT the major portion of the exhaust noise was due to shell radiation from the twin mufflers. These were mounted on the hood behind the operator. Any of several "wrapped" mufflers evaluated controlled the radiated noise but the airborne noise problem was more difficult to solve.

Only dual (twin mufflers) exhaust systems were considered. Initially, two "off-the-shelf" units reported to have performed well in DOT testing were obtained. One was from Stemco Manufacturing Company, Inc. and had about 1.7 times the volume of the original muffler. A second, having 2.2 times the volume of the original muffler, was from Riker Manufacturing, Inc. Airborne noise was measured near the source; 6" from the left tailpipe. Figure 3.4.1 shows the results. Both mufflers reduced noise above 500 Hz but performed worse than the original muffler at low frequencies.

Nelson Muffler Company agreed to design a muffler to give maximum attenuation using all available space on the hood. This muffler was about 2.5 times the volume of the original unit. Figure 3.4.2 shows that it had excellent low frequency attenuation but marginal mid-frequency performance to meet the overall goal. Nelson re-designed the internal baffling of the muffler and there was a distinct improvement in the 250 and 500 Hz octave bands, but not at higher frequencies (Second Nelson Muffler, Figure 3.4.2). Nelson was reluctant to use any absorptive lining in these units. If that were done the high frequency performance would likely be improved.

Measurements at the operator's ear indicated that overall, exhaust source strength had been reduced from 97 to 78 dB(A), and this was considered to be an acceptable level.

#### Engine Isolation Tests:

Although the neoprene machinery mounting pads seemed adequate to isolate mid and high frequency vibration, it was felt that some investigation of a more sophisticated mounting should be made. It was hoped this would improve low frequency isolation in particular.

There was adequate room at the front engine mounting points to incorporate two Lord Kinematic Center Bonded Mounts. Special upper and lower brackets were made to install these at a 45° angle to try to control transverse as well as vertical vibrational modes.

The fourth curve on Figure VI shows the result in terms of vertical acceleration levels in the frame near the right mounting point. The Lord mount is superior at high frequencies except for 1000 Hz. It is only slightly better at low frequencies. Measurements made on the two orthogonal axes at this point showed the Lord mount to be inferior to the simple neoprene pads.

The reason for the poor performance of the Center Bonded Mounts is not readily apparent. It should be noted that they were selected assuming a perfectly rigid vehicle frame and this was certainly not actually the case based on other vibration measurements made.

#### Phase IV: Final 90 and 85 dB(A) Kit Development

Tests made during Phase III indicated that a partial enclosure or operator shield of about three foot height behind and to the right of the driver could reduce sound levels by an average of 5 dB(A) at High Idle, Torque Converter Stall and High Lift conditions. As the program evolved, several other controls planned for the 85 dB(A) package had to be included in the 90 dB(A) package to drive the noise level down. The end result was that the only difference between the two kits was the operator shield. Because it is limited to the area to the rear and the right, this shield interferes with none of the normal operator functions and is not restrictive of visibility or access to the operator station. Figure 4.1.1 is a summary description of the components of the 90 dB(A) kit. Figure 4.1.2 describes the operator shield in more detail.<sup>8</sup>

#### Final Lab Tests of the 90 and 85 dB(A) Kits:

Figure 4.2 shows final lab test results with each of the two noise reduction packages under the operating condition of High Idle. Levels at the operator ear were reduced from 100.5 to 90 and 86 dB(A) respectively. The sole difference between the 90 and 85 dB(A) packages was the operator shield.

Figure 4.3 shows the effect of the two kits under the Torque Converter Stall condition. For the vehicle as received the noise spectrum was sharply peaked to 1000 Hz. This peak was caused primarily by the blade pass frequencies of the three elements of the torque converter. The overall level as received in the lab was 102.5 dB(A). The noise reduction kits lowered this level to 92 and 85 dB(A) respectively.

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<sup>8</sup> Complete detail and assembly drawings and parts lists were made and delivered to MERDC at the completion of the program.

Figure 4.4 shows the noise reduction achieved under the condition of High Idle Lifting an 8000 Lb. Load. Perhaps because during this test the effectiveness of the enclosures at the front of the vehicle is greatly reduced as the boom rises, the kits were not as successful under this condition. The first kit reduced levels to 92.5 dB(A). The addition of the shield brought a further reduction to 86 dB(A) in the lab.

Figures 4.5, 4.6 and 4.7 show the results at 3 measurement positions other than operator ear. In the vehicle "as received" the highest noise levels were in the 500 and 1000 Hz octave bands. The sound absorption treatment for the engine compartment was designed to have the greatest effect at these frequencies. Figure 4.5 shows that this was quite successful leading to reductions of 7 1/2 and 6 dB respectively. Figure 4.6 shows reduction in exhaust sound 6 inches from the left tailpipe. The advantage of the large volume Nelson mufflers is clearly shown in reductions at low frequencies ranging up to 17 dB at 125 Hz. Figure 4.7 shows the reduction in noise 6 inches from the intake achieved by installation of the added Donaldson Silencer. Again, at the frequency bands of greatest interest for the intake, 500 and 1000 Hz, there is a very substantial reduction. It was also necessary to add a damping treatment to the outside of the air cleaner to reduce the excessive radiated noise from these surfaces. Figure V13 shows the reduction in acceleration levels achieved by this treatment.

#### Effects of Some Added Control on a 90 dB(A) Kit:

As shown by Figures 4.2, 4.3 and 4.4 the goal of 90 dB(A) under all operating conditions was missed by an average of 1.5 dB in the lab. To determine if any single problem was causing this some further tests were made on the 90 dB(A) kit equipped vehicle.

Figures 4.8 and 4.9 show the results of these additional tests. In figure 4.8 the top and the sides of the engine compartment where numerous small cracks remained around the exhaust pipes and where the side panels are hinged on the vehicle were sealed. This sealing produced a reduction of less than 1 dB(A). These surfaces were covered with a composite barrier material consisting of 1/4" urethane foam "decoupling" layer bonded to a 0.8 lb/ft<sup>2</sup> barrier layer to reduce the radiated noise from them. This did achieve a significant reduction from 89 1/2 to 87 1/2 dB(A). In order for this added mass to be effective, however, it might be necessary to first plug all the small cracks and leaks. This would probably require a totally new hood for the vehicle.



Installation of remote exhaust lines produced only a very marginal reduction in noise from 87.5 to 87 dB(A). Turning the fan off resulted in another reduction of 0.5 dB(A).

Finally, in addition to all the above mentioned controls Figure 4.9 shows the effect of removing the prop shaft from the vehicle. This stops the torque converter and both transmission-mounted hydraulic pumps. The change in noise levels at high idle is no longer very impressive. The engine compartment absorption treatment, flexible hydraulic suction lines, and transmission and reservoir isolators seem to be doing their jobs. (Under conditions of torque converter stall and high lift, however, the torque converter and lift pump respectively are still important noise sources.)

From these tests it seemed that any further significant reduction in noise at the operator ear of this vehicle without the use of operator enclosure would require improvement in virtually all of the noise control measures already taken. Design complexity and cost would increase substantially as a new hood and new engine side panel would be required.

#### Phase V: Final Field Data Acquisition

In May of 1974 the RTFT was returned to the Dana Corporation Technical Center for final performance and noise tests. Results are discussed in detail below but may be summarized as follows:

- (1) Neither kit degraded vehicle performance.
- (2) The 90 dB(A) Kit reduced noise at the operator's ear by an average 10.5 dB(A).
- (3) The 85 dB(A) Kit reduced noise at the operator's ear by an average 15.3 dB(A).
- (4) Spectator noise 50 feet away was reduced by an average 5 dB(A).

#### Final Performance Tests:

Figure 5.1 presents the final performance test results in the same format as the initial test, Figure 1.18. The initial test was made with no load on the forks. Possibly for this reason, our drawbar pull (tractive force) was limited by wheel slip to 8500 lbs. In the final test a 7000 lb. load was placed on the forks. With this

load a tractive force of 10,500 lbs. was achieved with a resulting increase in reserve tractive horsepower from 79 to 84 HP. Comparison of the vehicle temperatures in terms of °F above ambient ( $\Delta T$ ) shows very little change from the baseline condition; the relative top tank temperature changed only 1°F.

#### Final Noise Tests:

Figure 5.2 presents a comparison of noise levels at the operator's ear position. The kits were most successful for the high idle condition which was the condition used for most of the development work. They were least successful for the condition most dissimilar to the lab test conditions, that of Full Throttle Acceleration.

Some added measurements were made at various road speeds and these are also presented in Figure 5.2. The shield is rather ineffective at high road speeds. This may be due to drive train noise, tire noise, and conceivably even extra air turbulence at the operator's ear due to the shield. At moderate road speeds in intermediate gear the shield apparently works well with noise levels of about 85 dB(A).

Spectator noise at a distance of 50 feet is compared in Figure 5.3. Only the 90 dB(A) configuration is shown; the shield has negligible effect on spectator noise. Levels are highest to the rear of the 90 dB(A) machine and are almost certainly controlled primarily by the diesel engine.

ISO-SPL lines are shown on Figure 5.4. Calculation will show that, on average, the sound levels decrease at a rate of about 6 dB per doubling of the distance from the operator position indicating that it is a reasonable approximation of the "center" of the noise sources for this test. Original ISO-SPL lines were presented in Figure 1.10.

## GENERAL DISCUSSION

Specific data have been discussed in the various sections under Investigation. This section is limited to a brief discussion of some operational and technical problems which may be of general interest.

### Measurement and Analysis

The "window" technique of octave band measurement was used extensively and successfully in this program. Care must be taken, however in dealing with as complex a machine as the RTFT. For example, while the window technique was adequate to determine the overall importance of the hydraulic system to sound at the operator's ear, it was not adequate to specify the relative importance of different hydraulic components. This required narrow band analysis, some near-field noise measurements and numerous vibration measurements. The window technique is adequate for "sorting out" a group of parallel sources and/or paths, but is not sufficient where there are series combinations of sources and paths or when there are complex interactions among sources.

A constant 6% bandwidth narrow band analyzer was used for spectrum analysis. In some cases this unit did not have adequate frequency resolution to quickly do the job required of it because of the large number of sources operating at various and only slightly different multiples of engine speed. A 500 line real-time narrow band analyzer would have been a most useful tool in studying the RTFT.

At the beginning of the program a very large number of acceleration (vibration) measurement positions were investigated. A small number of these which seemed most informative were then selected for continued measurement throughout. This approach worked well. Occasionally points other than those continually monitored were compared to the baseline measurement as a check on the progress being made.

### Vehicle Malfunctions

Several times during the project there occurred a vehicle malfunction or failure. Each such event had a possible effect on acoustical noise. Some, such as an exhaust pipe leak, were rather easily discovered and corrected. Others, such as the progressive failure of the transmission oil charging pump, were not at all obvious and

temporarily led to erratic noise data. This type of problem can probably be minimized if the noise test personnel fully understand the function of the machine and monitor its important operating parameters, but the most important precaution may be to try to check vehicle performance and function immediately if spurious data seem to appear.

#### Field vs. Lab Testing

The RTFT operates much of the time in a stationary position or while moving at a low rate of ground speed. Laboratory stationary tests simulate these conditions well. The vehicle is, however, also capable of traveling at a considerable ground speed of nearly 30 miles per hour. As has been discussed in the Phase V section of Investigation, the treatments developed based on stationary tests were not as effective at high vehicle speeds. Perhaps if the vehicle had been returned to the test track for moving tests at several points in the program this could have been partially avoided. To have done so would have increased development costs.

#### Cost/Performance Trade-offs

A number of laboratory treatments such as underside enclosures were shown by the data to have had little effect compared to the cost and/or difficulty of implementing them. Thus, no prototype parts were designed or fabricated. In light of the fact that the noise goals were not completely met, perhaps more of these treatments should have been fabricated and tried out.

#### Prototype Part Procurement

An attempt was made in this program to fully define each noise problem area before procuring prototype parts for that problem. The seemingly ever-lengthening lead times for prototype hardware were a major factor in a two month longer than planned project time. It appears that wherever possible, long lead time parts must be ordered based partly on experience and intuition even though the acoustical noise requirements are not fully defined, if reasonable development schedules are to be maintained.

## CONCLUSIONS

A great amount of data has been presented in this report. It represents a lengthy and involved investigation of how noise is generated and propagated in the Rough Terrain Forklift Truck. It has resulted in the design of specific noise reduction treatments for this specific vehicle. Two questions immediately present themselves. First, what information of general interest was gained in this project? Second, how can this information be applied to noise reduction of other vehicles?

We feel the following points are important and of general interest.

1. The noise of a complex vehicle such as the RTFT may not be significantly reduced using only off-the-shelf components. This is because the initial design will generally not have taken into account the need for properly fitting enclosures, component isolation or acoustical materials treatments.
2. The program has demonstrated that noise reductions of 10-15 dB(A) are feasible without the use of "exotic" materials or development techniques. Total cost of the 85 dB(A) Kit including labor ought to be less than 5% of present vehicle cost. Performance of the RTFT was not degraded.
3. Future programs should reverse the sequence of development events of this program. The maximum noise reduction kit should be developed first. In doing this the noise "system" will be fully defined. Then lesser noise reduction kits can be configured from the most cost-effective and readily available components of the major kit.
4. Techniques of engine intake and exhaust silencing, isolation, damping, absorption and barrier enclosures are all fairly readily available. The greatest difficulty experienced was in trying to reduce the noise due to the hydraulic system. A major problem was that this system readily transferred large amounts of energy among a number of noise generating elements located throughout the vehicle.

Returning to the second question posed above, how can this information be applied to other equipment? In terms of project organization, the investigator should undertake the largest noise control effort first. For most types of problem, he may be fairly confident that existing technology, properly applied, can solve his problems successfully.

When the problem involves a complex distributed system of inter-related noise sources such as the RTFT hydraulic system, extreme caution is required. If at all feasible, a way should be found to study this system independently of other noise sources. (In the RTFT, for example, "silent" electric motors might be used to drive the gear pumps so that effects of engine, transmission, etc. would be totally eliminated.)

### RECOMMENDATIONS

The experience of the noise reduction effort on the RTFT would seem to indicate a need for further research into hydraulic system noise. The U.S. Army MERDC has supported work to measure the airborne noise of hydraulic components. A very useful next step might be to make comprehensive noise and vibration measurements on an existing typical hydraulic system under all operating parameters. This would show the relative importance of pumps, valves, lines, cylinders and the inter-relationships among them. Recall, for example that both alternative power steering pumps evaluated in the RTFT appeared equally successful in reducing reservoir vibration when operated alone. When operated with the other two pumps, however, one of them was clearly superior.

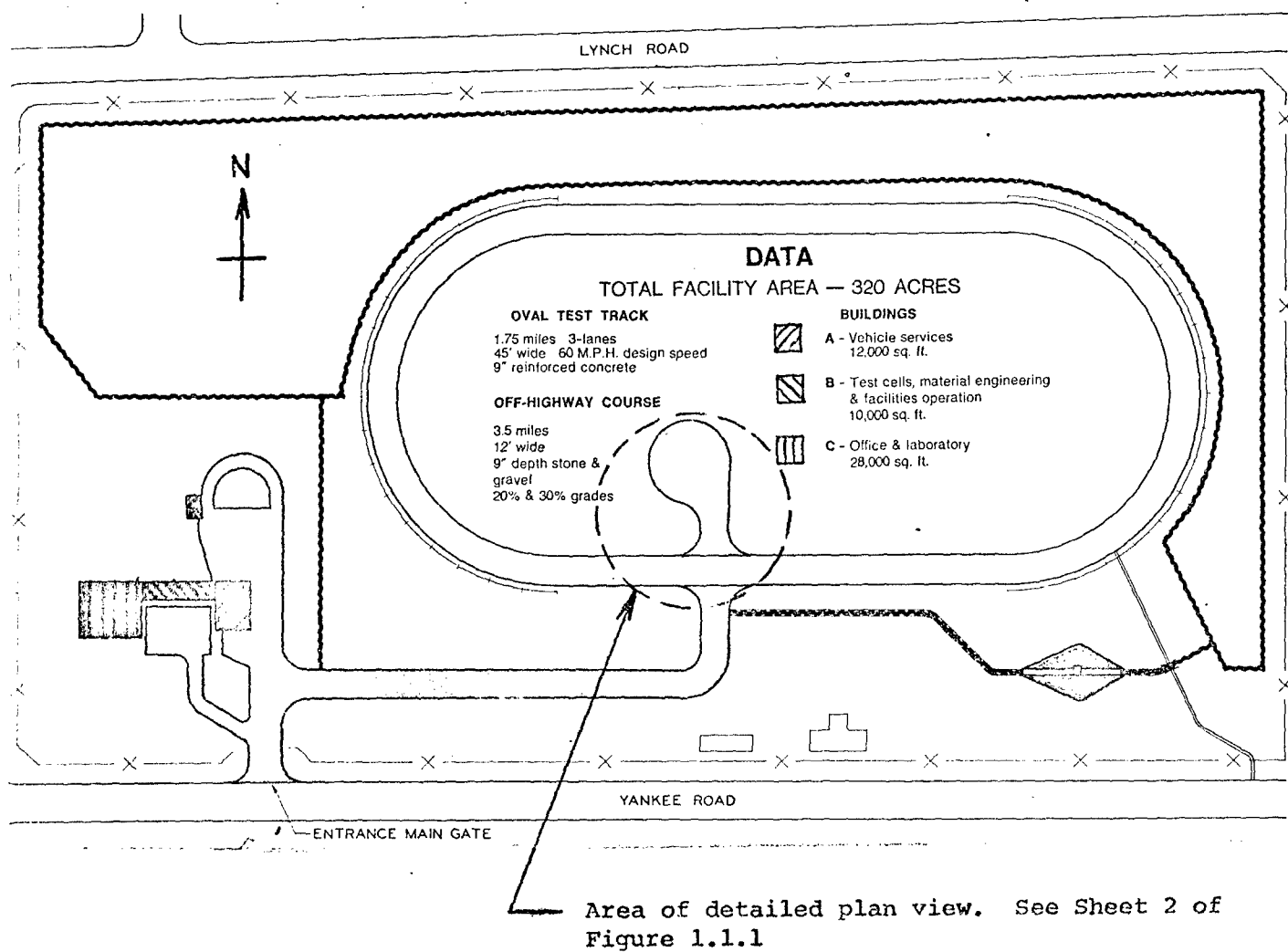


FIGURE 1.1.1 (Sheet 1 of 2)

PLAN VIEW OF THE DANA CORPORATION TECHNICAL CENTER AND TEST TRACK

Access drive to track and turn-around pad where Rough Terrain Forklift Truck stationary measurements were made are asphalt paved.



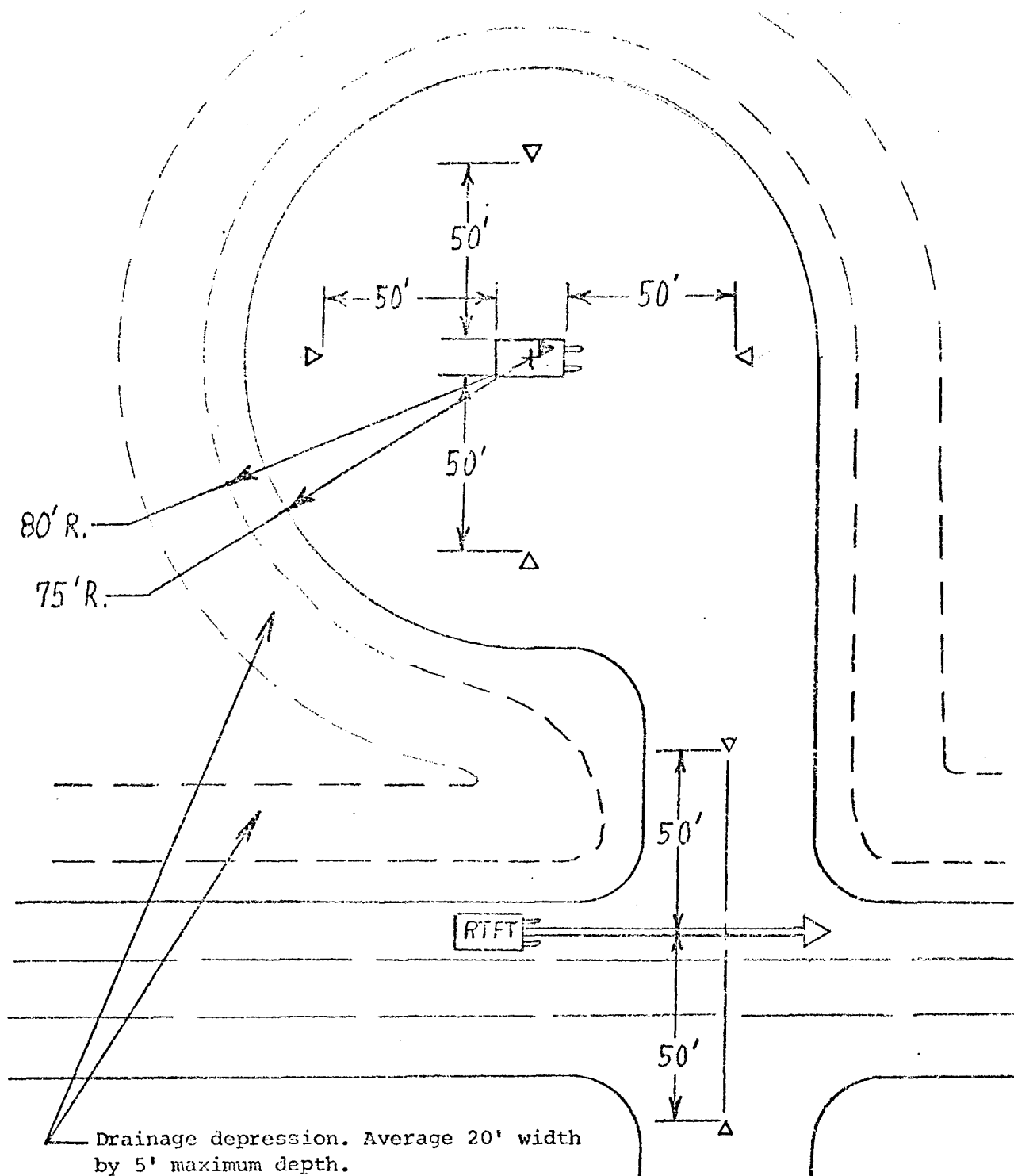


FIGURE 1.1.1 (Sheet 2 of 2)

DETAILED PLAN VIEW OF TEST SITE FOR STATIONARY AND MOVING TESTS OF THE 10,000 LB. ROUGH TERRAIN FORKlift TRUCK

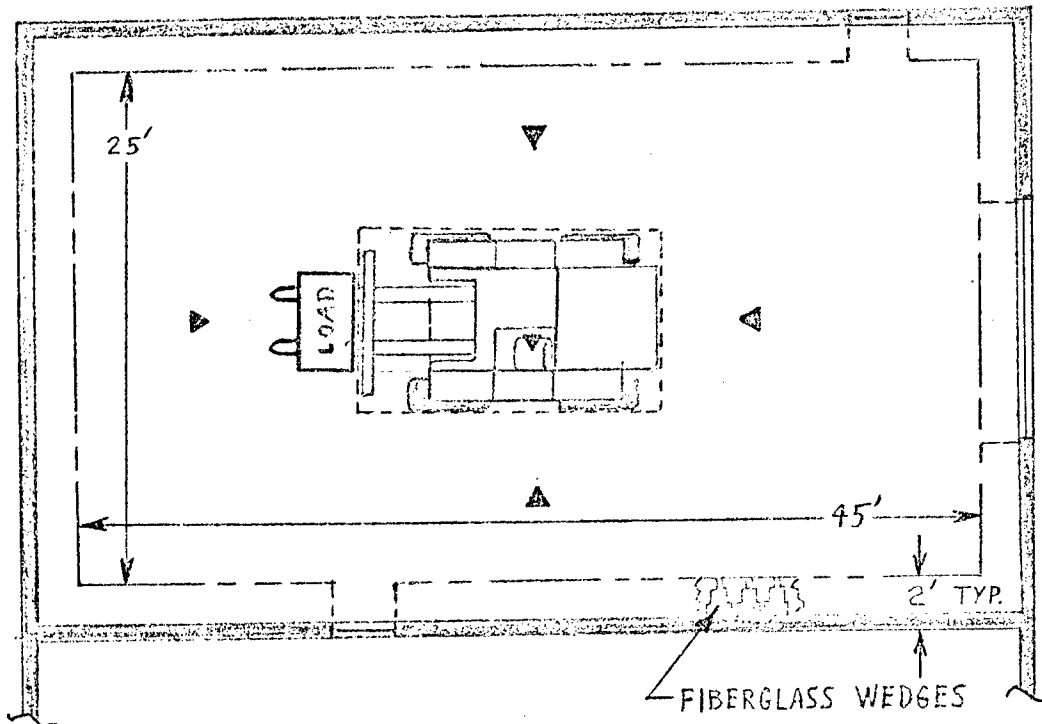
Concrete track and flat asphalt turn-around pad surrounded by short grass.

Operator's ear microphone position - 6" from right ear.  
Spectator noise microphone position - 4' above pavement.

FIGURE 1.1.2

LAB TEST SET-UP FOR THE RTFT

▲ Denotes microphone position



The walls and ceiling of the semi-anechoic room are covered with highly absorptive fiberglass wedges. The floor is poured concrete. The vehicle was centered in the room. The microphone positions shown are operator ear location and five feet high, four feet from surface of vehicle or load at rear, sides and front.

FIGURE 1.2

SOUND AT THE OPERATOR'S EAR POSITION-STATIONARY

- ▽ Idle
- High Idle
- △ Torque Converter Stall
- High Idle-Raising Maximum Load

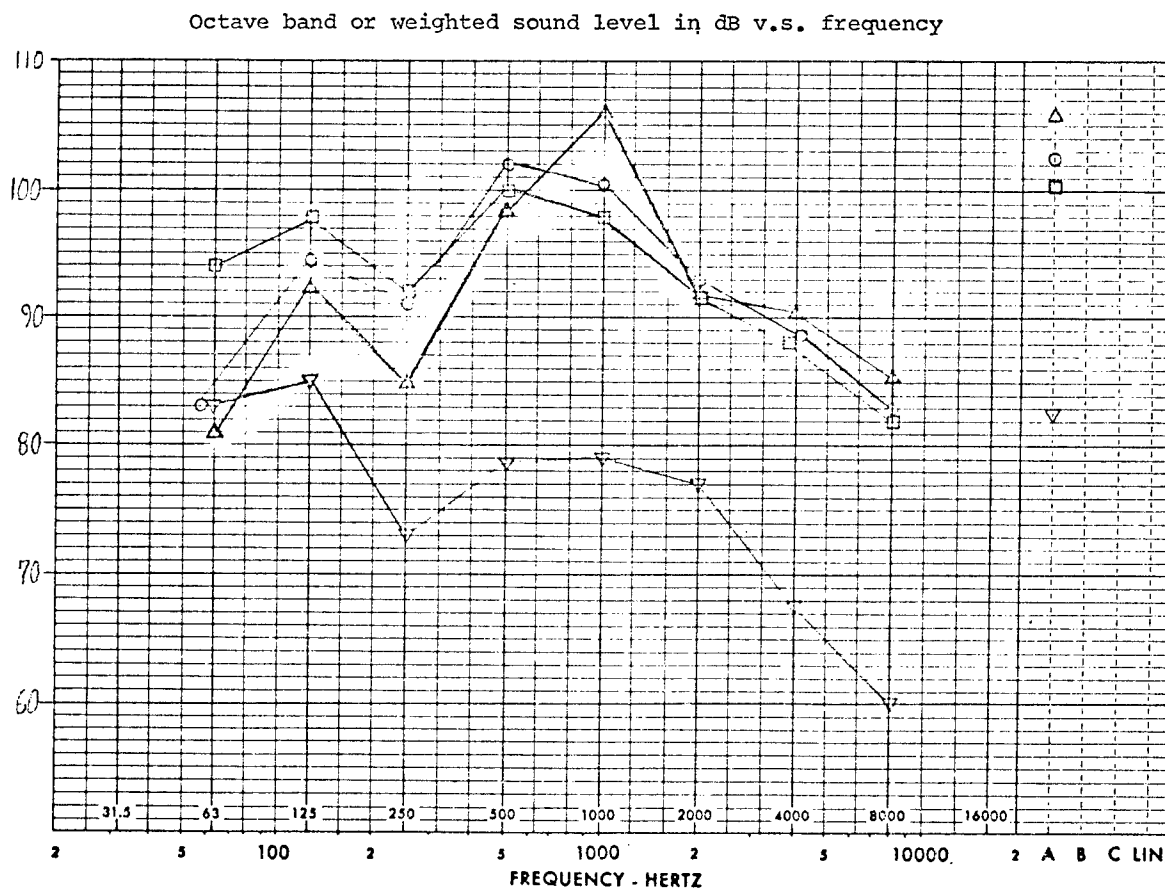


FIGURE 1.3

SOUND 50 FEET TO THE FRONT-STATIONARY

- ▽—▽ Idle
- High Idle
- △—△ Torque Converter Stall
- High Idle Raising Maximum Load

Octave band or weighted sound level in dB v.s. frequency

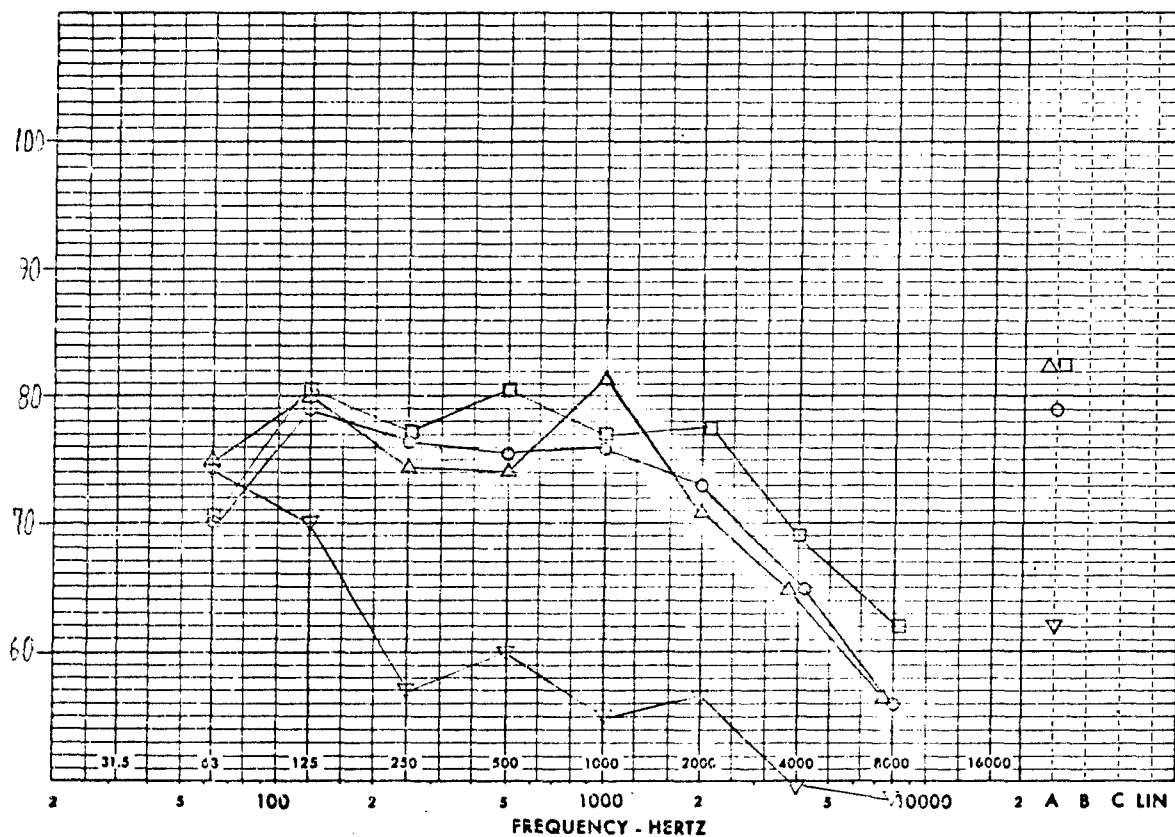


FIGURE 1.4

SOUND 50 FEET TO THE LEFT SIDE-STATIONARY

- ▽——▽ Idle
- High Idle
- △——△ Torque converter stall
- High Idle Lifting Maximum Load

Octave band or weighted sound level in dB v.s. frequency

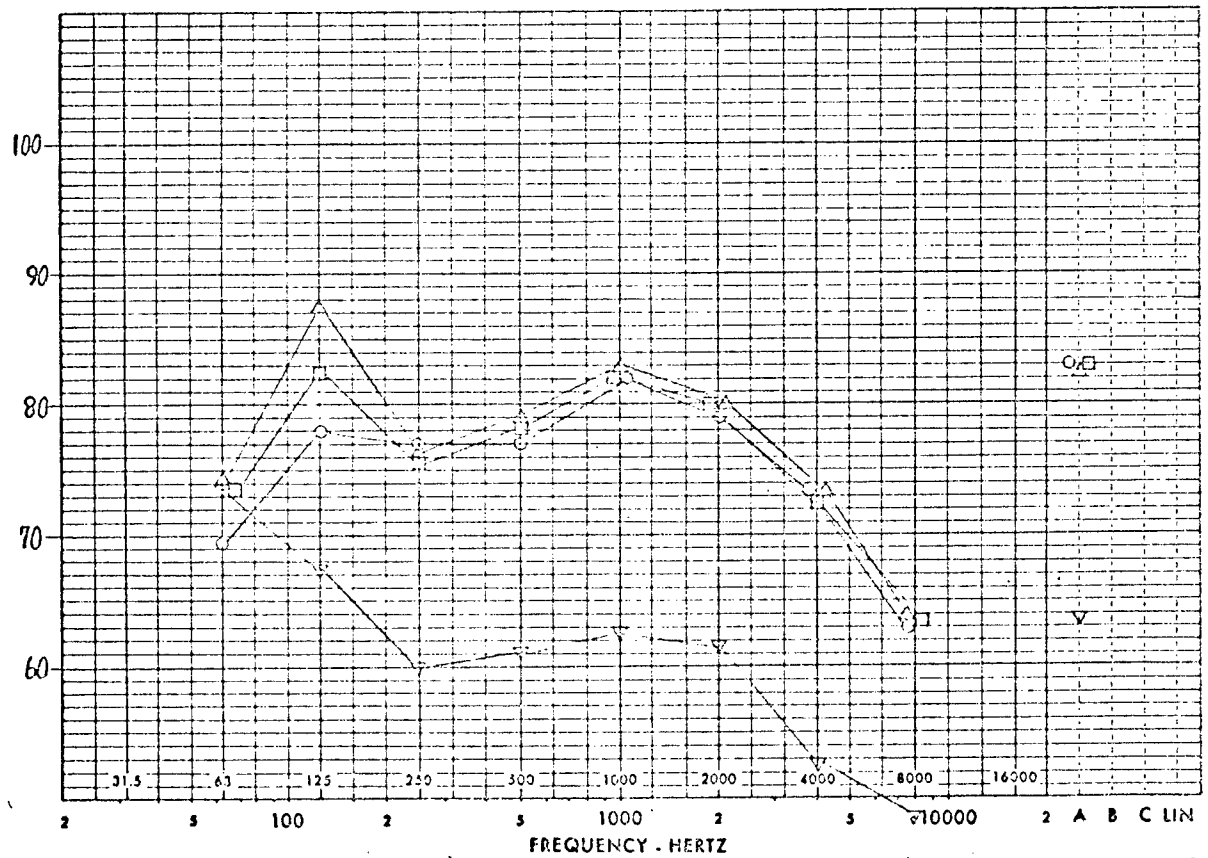


FIGURE 1.5

SOUND 50 FEET TO THE REAR-STATIONARY

- ▽——▽ Idle  
 ○——○ High Idle  
 △——△ Torque Converter Stall  
 □——□ High Idle Lifting Maximum Load

Octave band or weighted sound level in dB v.s. frequency

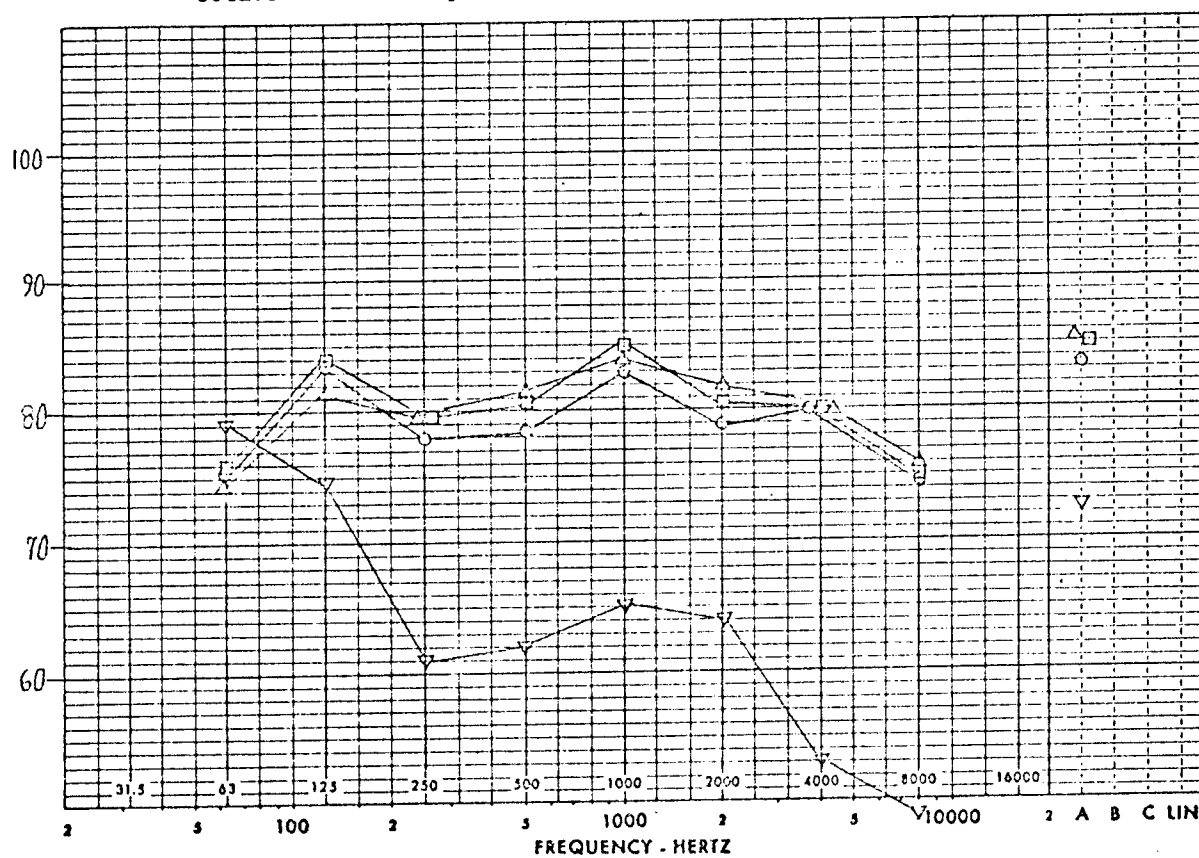


FIGURE 1.6

SOUND 50 FEET TO THE RIGHT SIDE-STATIONARY

- ▽ Idle
- High Idle
- △ Torque Converter Stall
- High Idle Lifting Maximum Load

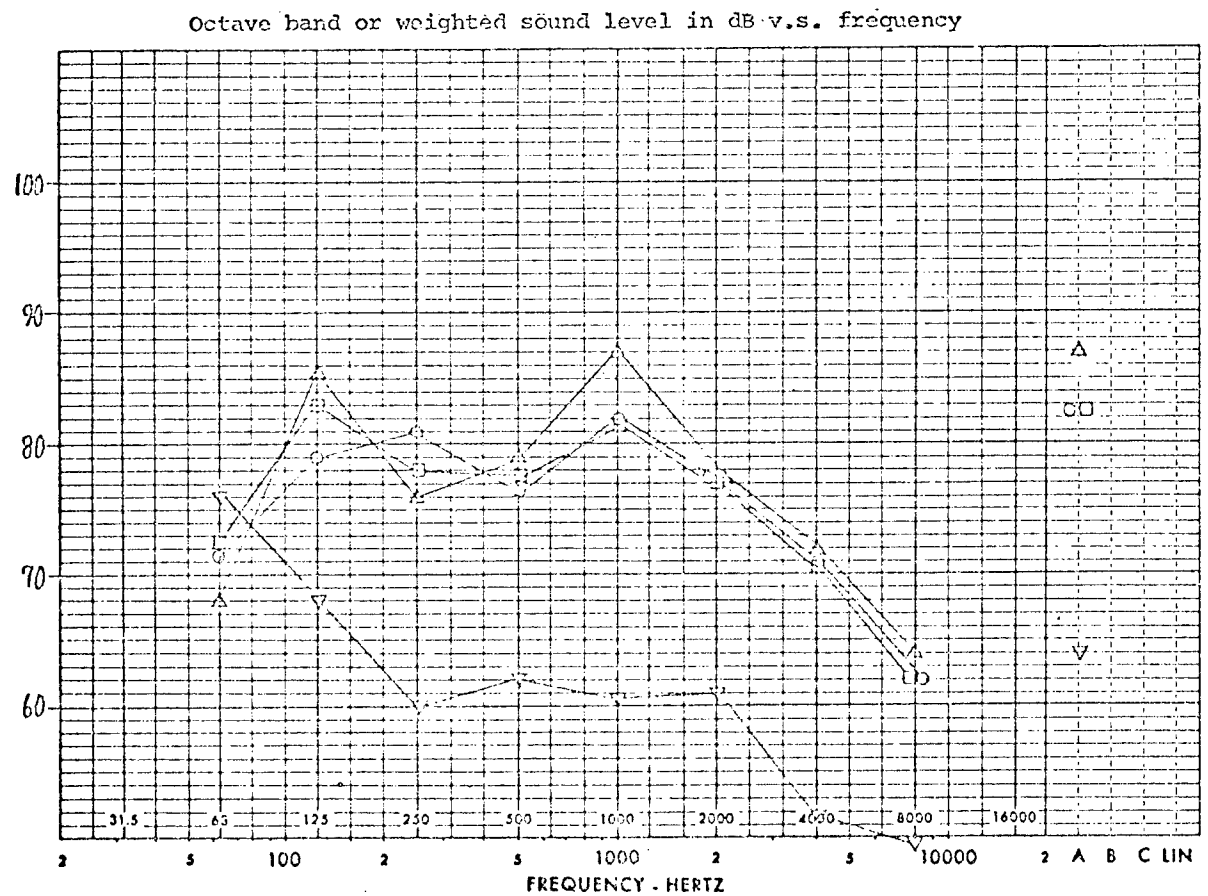


FIGURE 1.7

SOUND AT THE OPERATOR'S EAR POSITION-MOVING

- ◇ Intermediate Gear Loaded to Rated Speed
- ◆ Full Throttle Acceleration

Octave band or weighted sound level in dB v.s. frequency

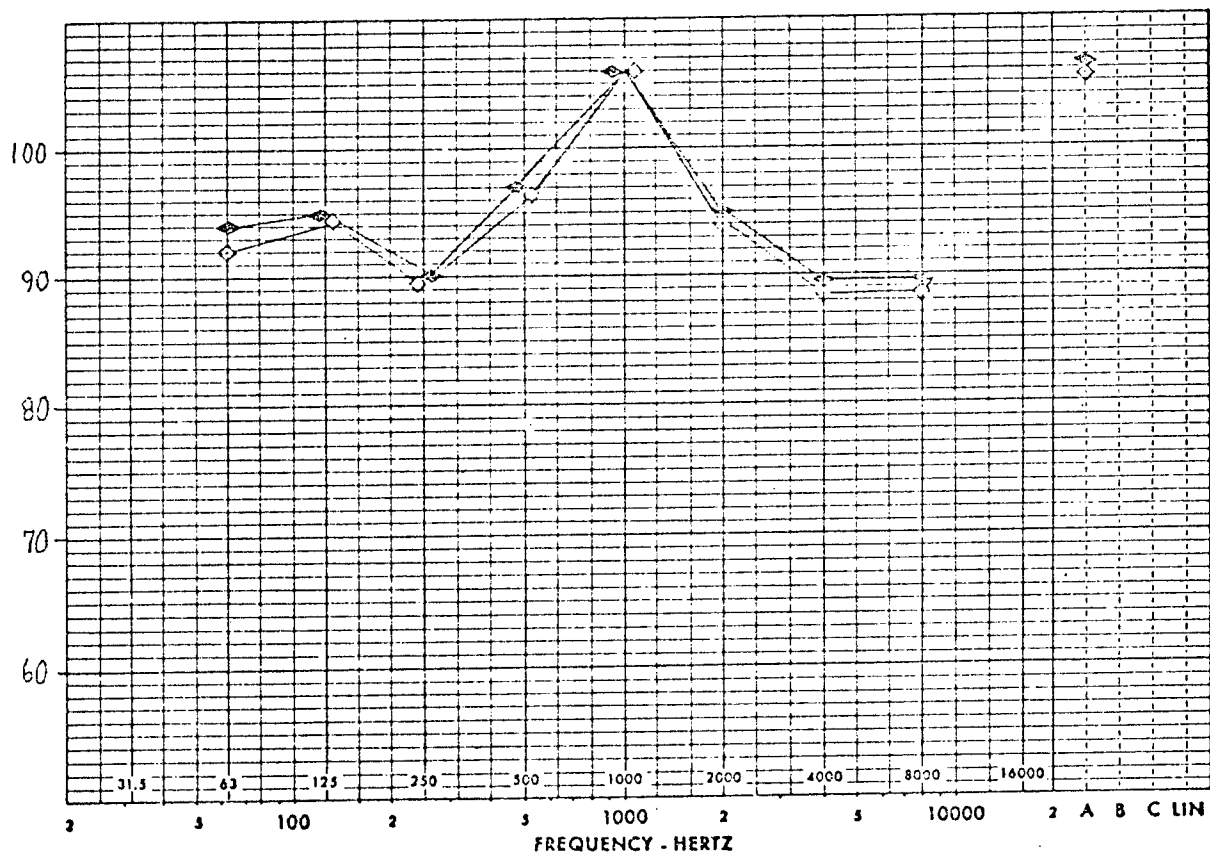




FIGURE 1.8

SOUND AT THE RIGHT SIDE PASS-BY POSITION

- ◇ Intermediate Gear Loaded to Rated Speed  
◁ Maximum Vehicle Speed-No Load

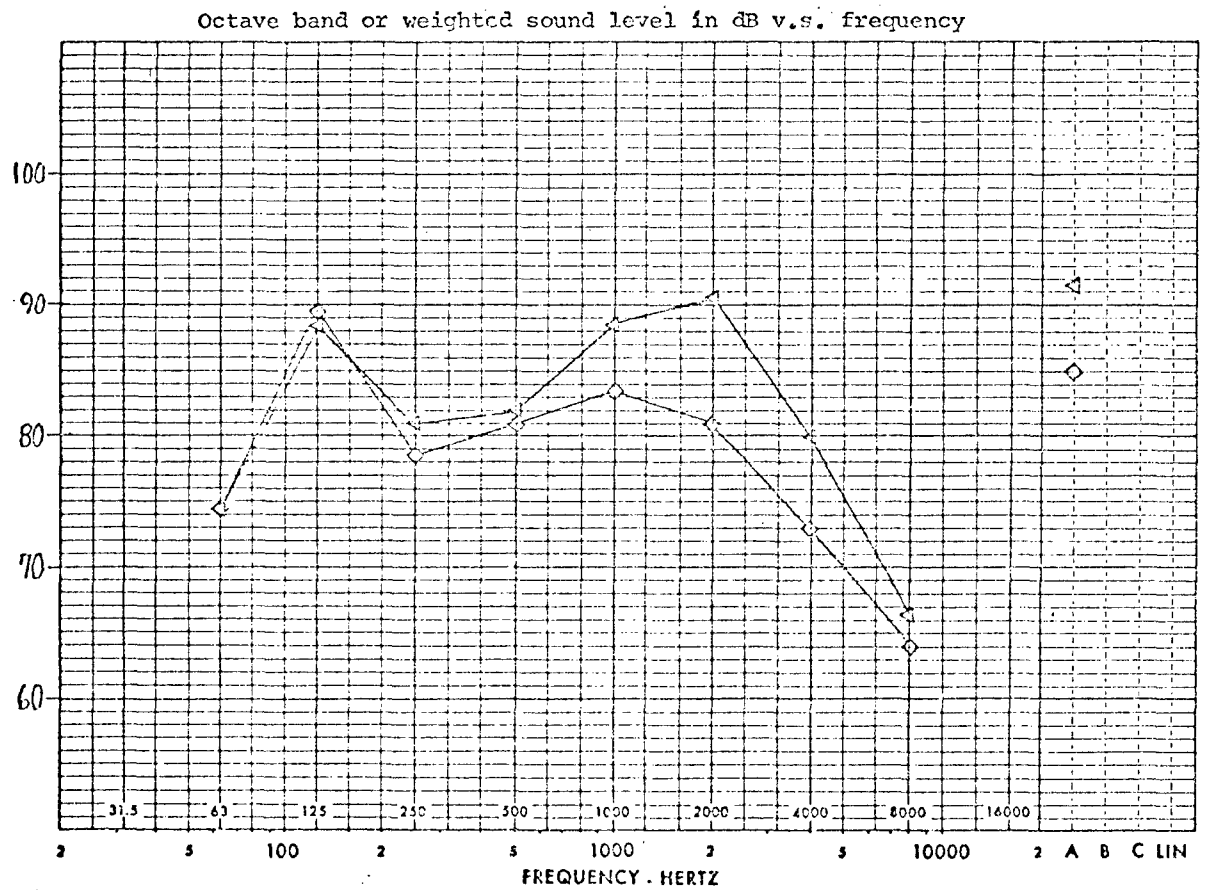


FIGURE 1.9

SOUND AT THE LEFT SIDE PASS-BY POSITION

◇ Intermediate Gear Loaded to Rated Speed  
 ◁ Maximum Vehicle Speed-No Load

Octave band or weighted sound level in dB v.s. frequency

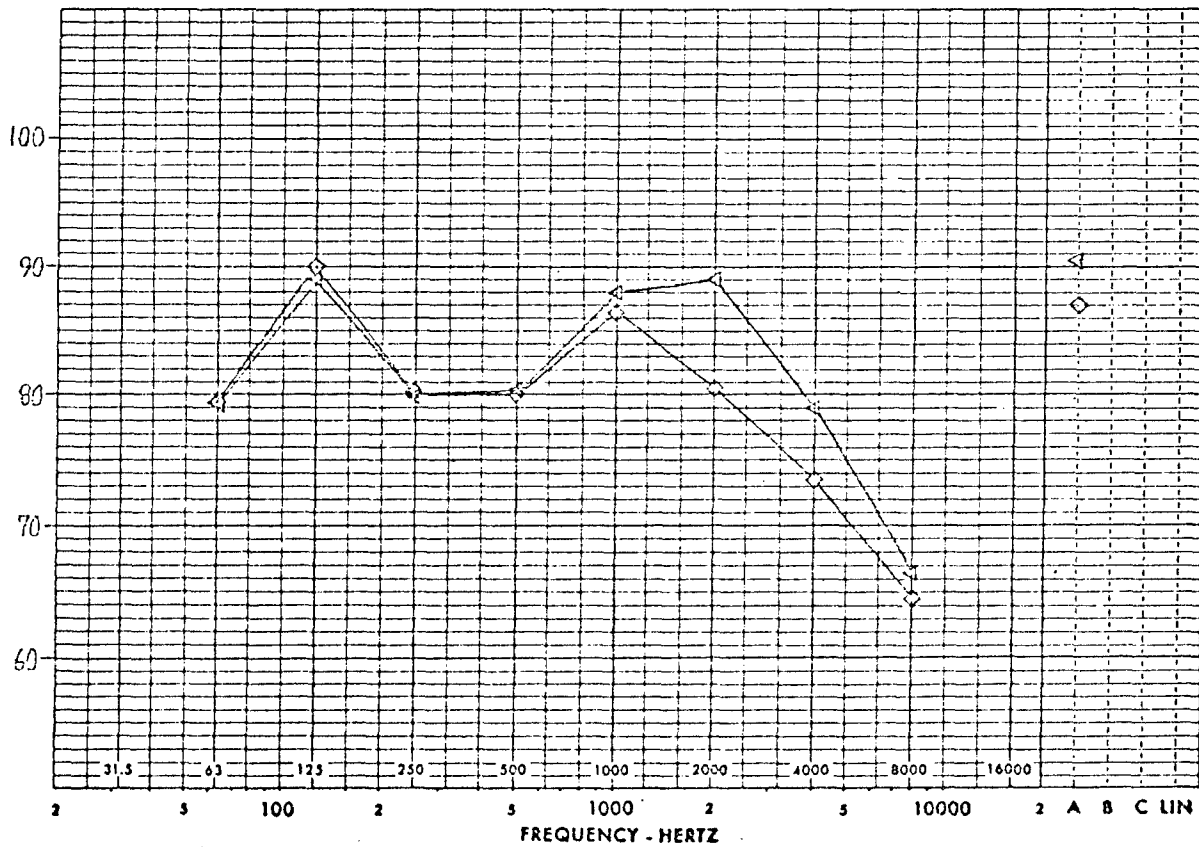
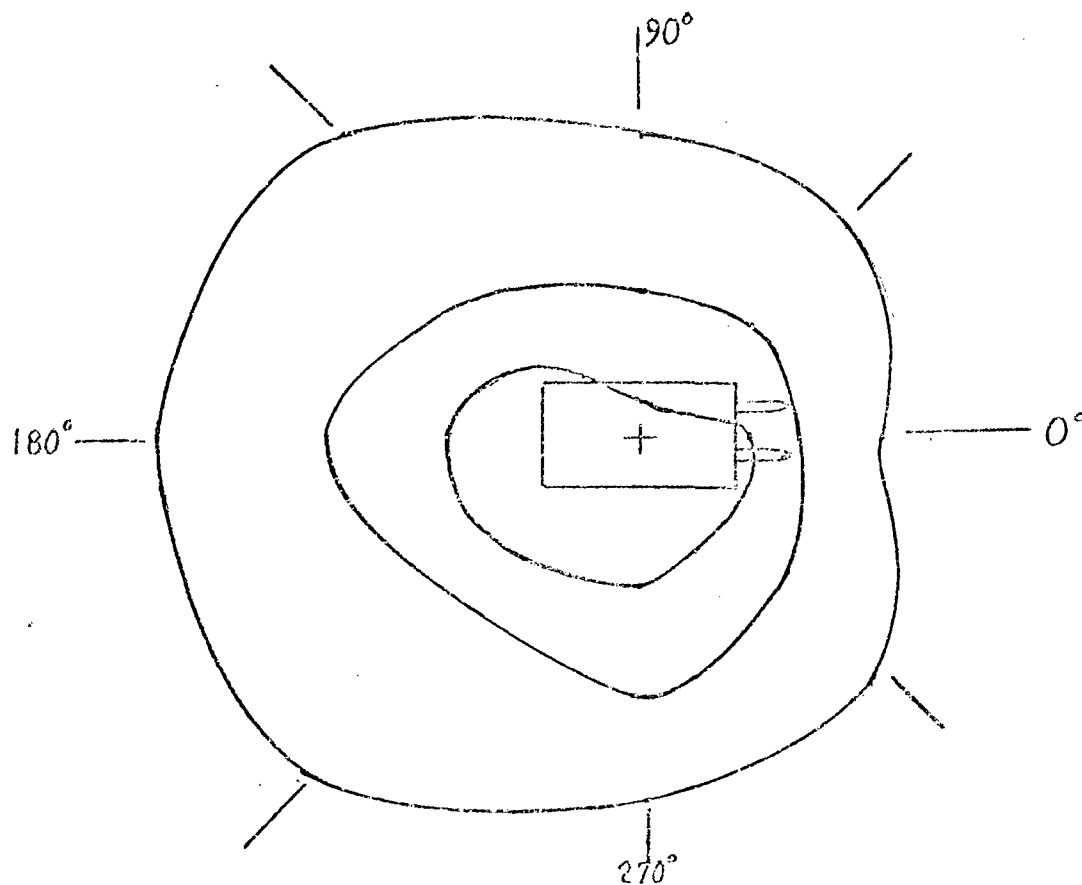


FIGURE 1.10

ORIGINAL ISO-SPL LINES FOR HIGH IDLE CONDITION

Origin of axes at operator's ear which approximates center of vehicle.  
Level at origin was 103.5 dB(A). Measurements made with handheld Bruel and Kjaer Type 2206 Sound Level Meter with Type 4148 free field microphone and windscreen. Meter on A-slow setting held at ear height of standing observer.



$\Delta$ dB (A)	DISTANCE FROM OPERATOR EAR ALONG AXIS (FT)							
	0°	45°	90°	135°	180°	225°	270°	315°
-5	9	3	3	7	16	14	13	10
-10	13	13	12	17	26	21	21	17
-15	20	25	25	35	40	39	30	28

FIGURE 1.11

SOUND AT OPERATOR'S EAR POSITION

- ▽ Idle
- ◇ 1/2 High Idle Engine Speed
- High Idle
- △ Torque Converter Stall
- High Idle - Lifting 8000 Lb. Load

Octave band or weighted sound level in dB v.s. frequency

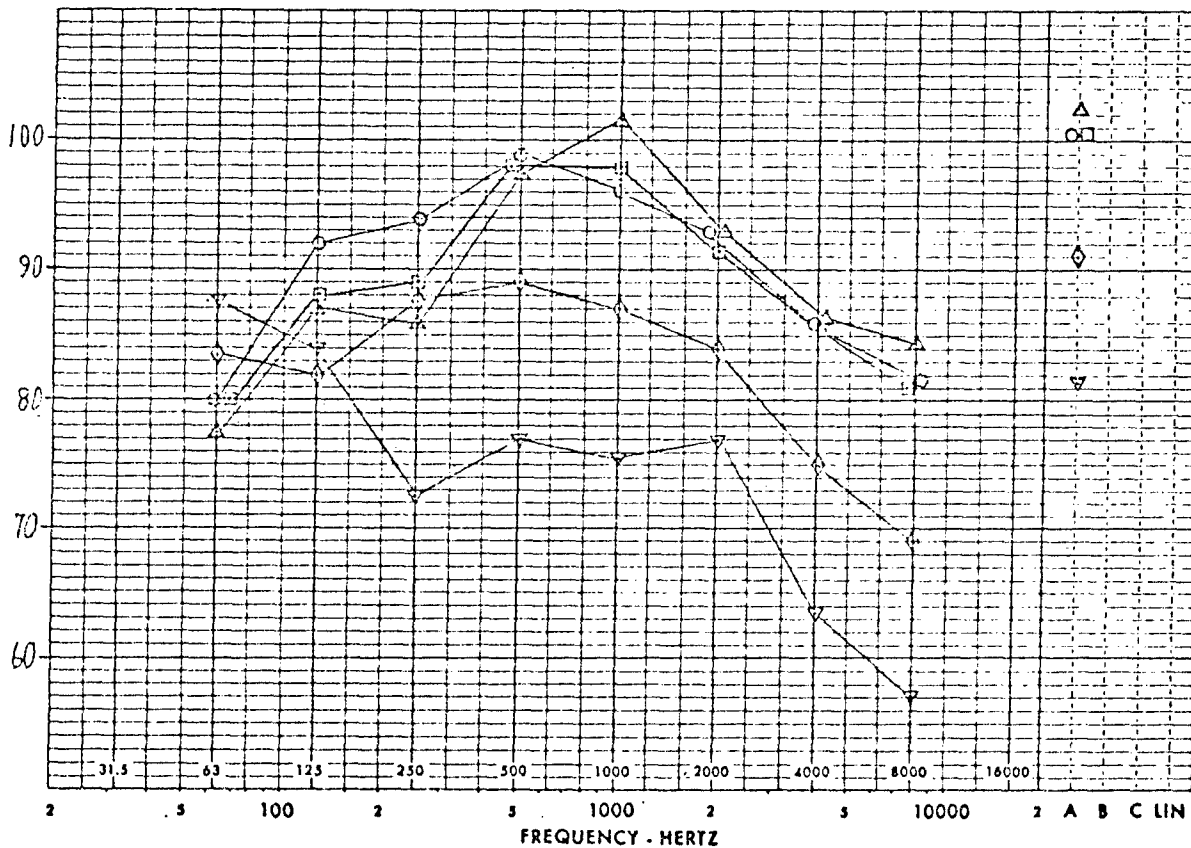


FIGURE 1.12

SOUND 4 FEET FROM THE RTFT -HIGH IDLE CONDITION

- ①—① 4 feet from front of load (7.5 feet from front)
- ②—② 4 feet from left side
- ③—③ 4 feet from right side
- ④—④ 4 feet from rear

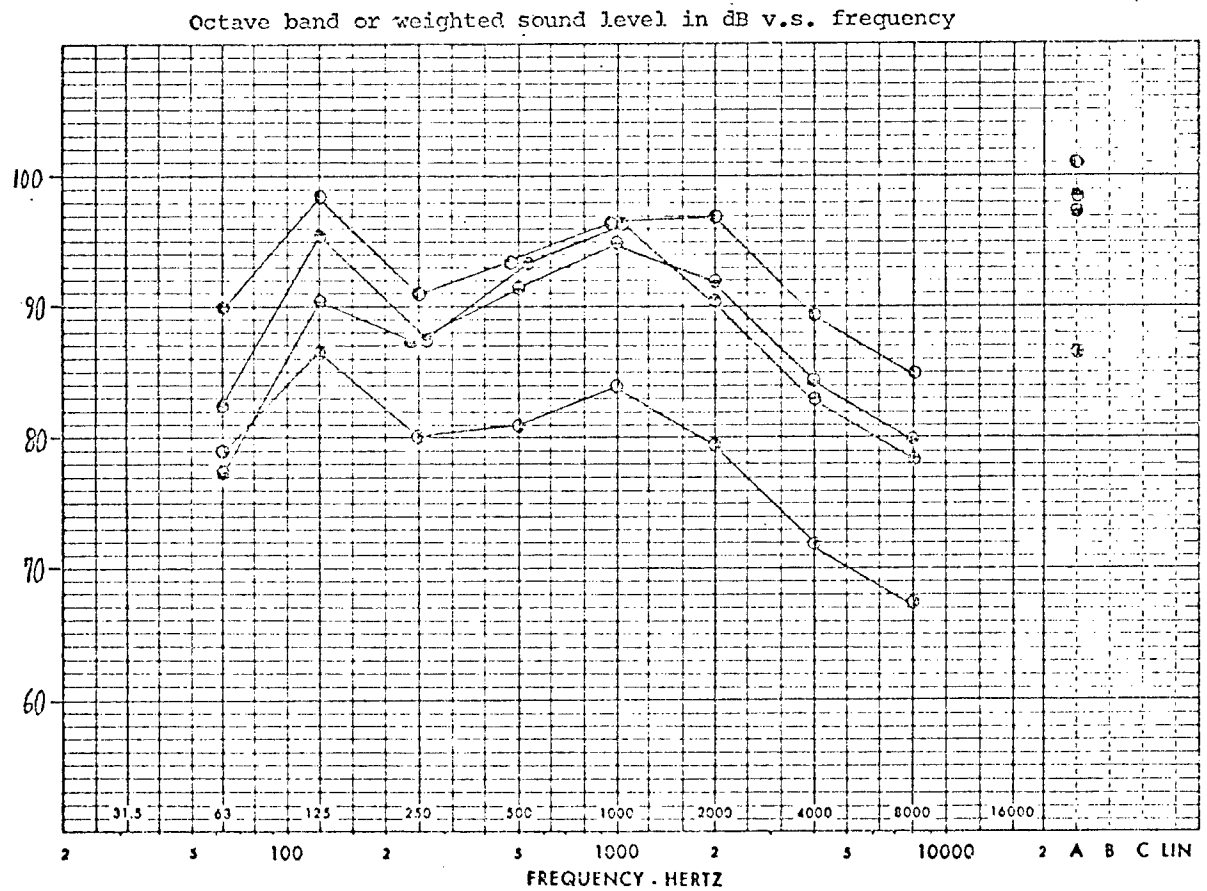


FIGURE 1.13  
SOUND IN ENGINE COMPARTMENT

- ▽ Idle
- High Idle
- △ Torque Converter Stall
- ◇ 1/2 High Idle Engine Speed
- High Idle - Lifting 8000 Lb. Load

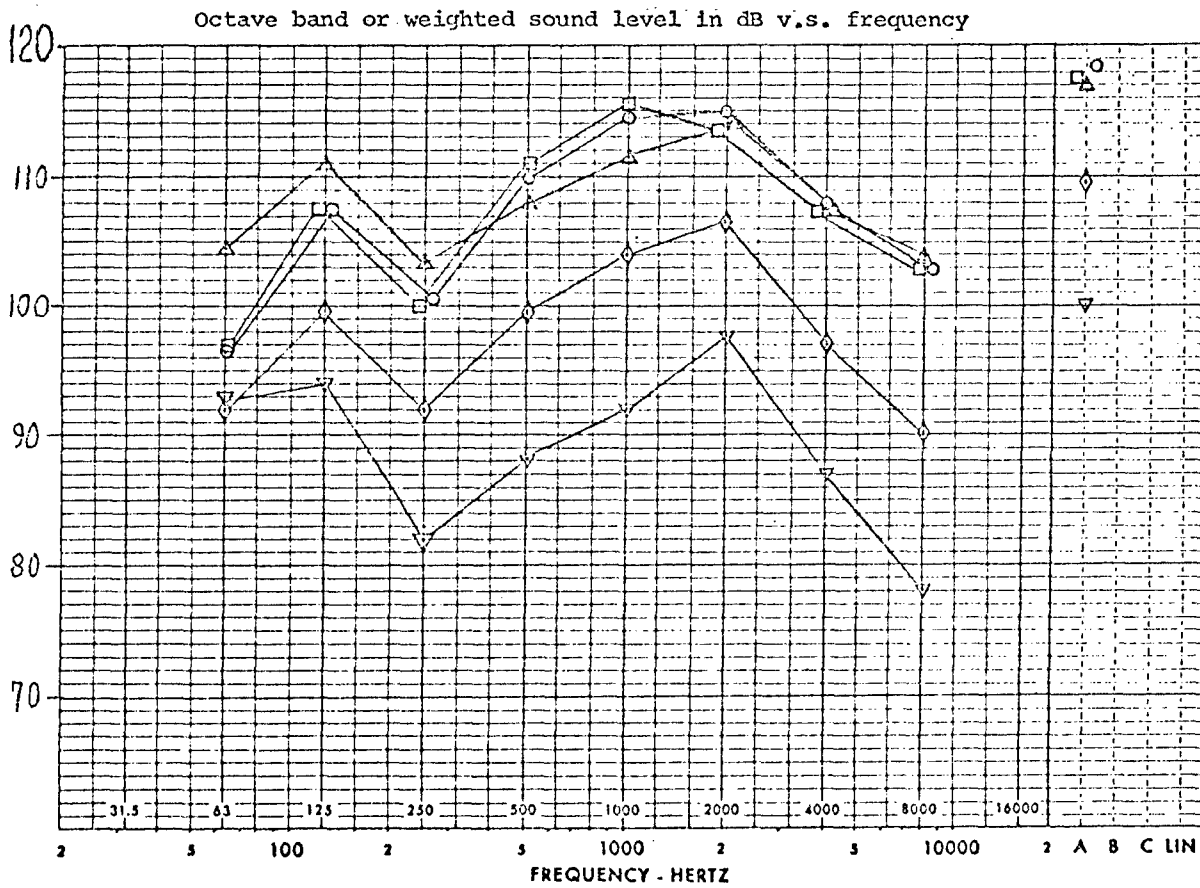


FIGURE 1.14

SOUND IN BODY UNDER OPERATOR

- ▽ Idle
- High Idle
- △ Torque Converter Stall
- High Idle-Lifting 8000 Lb. Load
- ◇ 1/2 High Idle Engine Speed

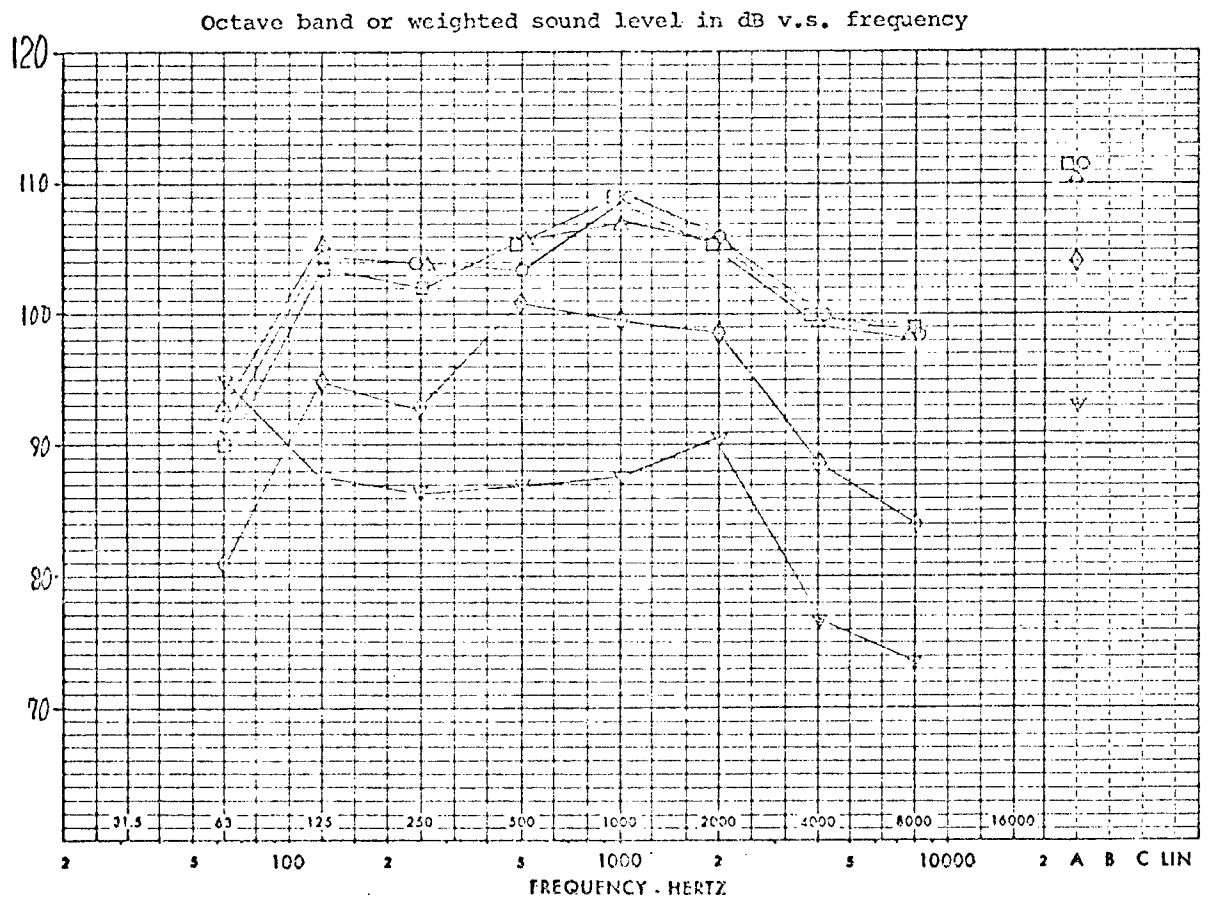


FIGURE 1.15

SOUND 6 INCHES FROM EXHAUST - HIGH IDLE CONDITION  
AND 1/2 HIGH IDLE CONDITION

- 6" from right exhaust - High Idle
- 6" from left exhaust - High Idle
- 6" from right exhaust - 1/2 High Idle
- 6" from left exhaust - 1/2 High Idle

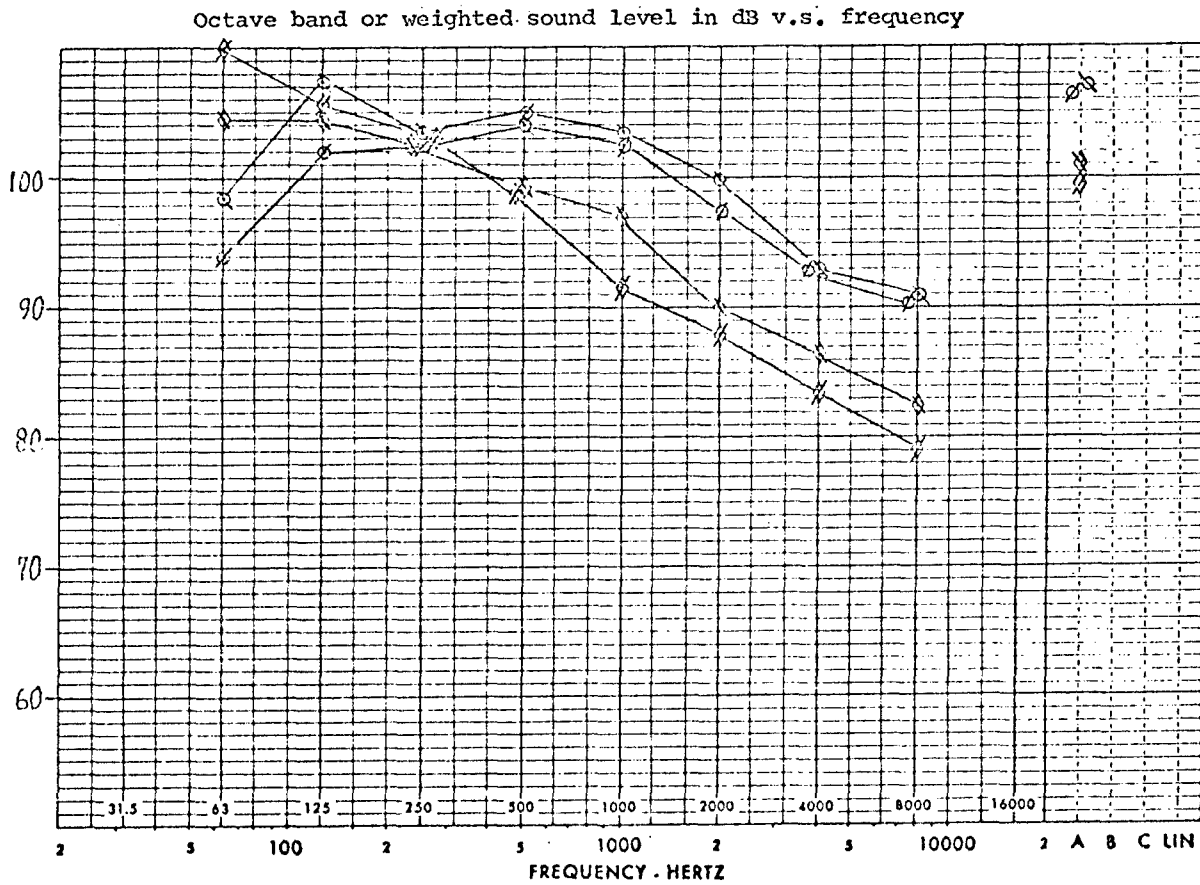




FIGURE 1.16

SOUND 6 INCHES FROM RADIATOR CENTER

Microphone on rotational axis of fan 6" to rear of vehicle from radiator screen

○ — High Idle  
◇ — 1/2 High Idle engine speed

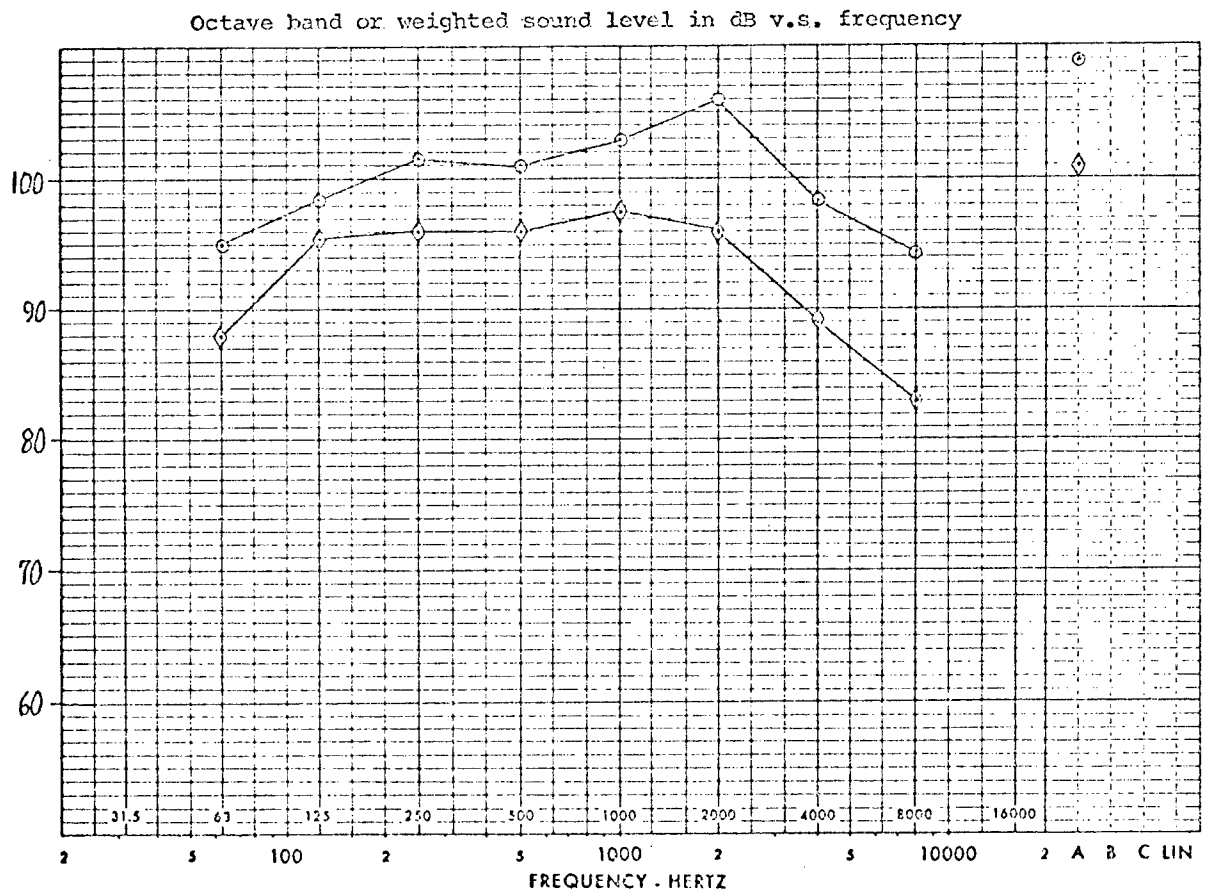


FIGURE 1.17

SOUND 6 INCHES FROM AIR INTAKE

○ High Idle  
◇ 1/2 High Idle engine speed

Octave band or weighted sound level in dB v.s. frequency

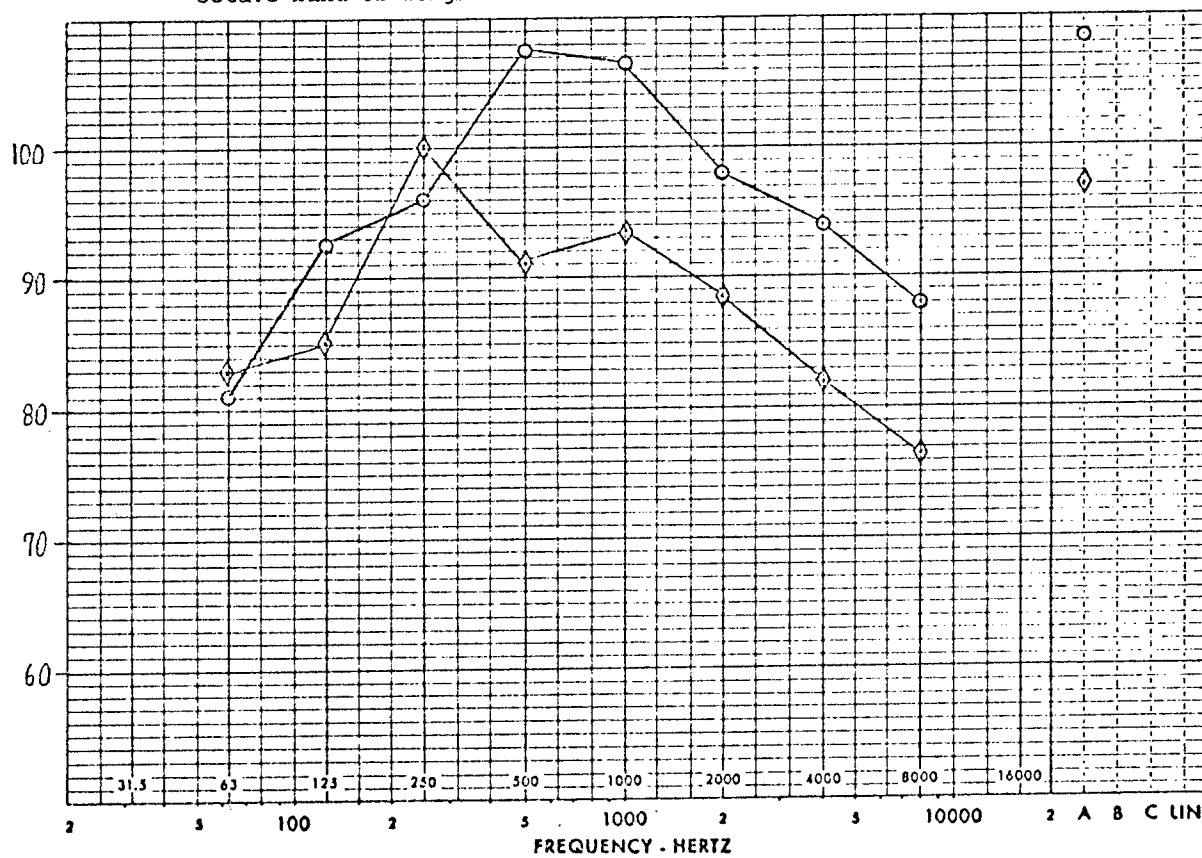


FIGURE 1.18

PERFORMANCE TEST DATA  
U.S. ARMY ROUGH TERRAIN FORK-LIFT TRUCK

Operation in low gear with four-wheel drive and two-wheel steering. Stable conditions per SAE J872a and SAE J819a. No load on forks.

Date	15 Aug. 1973
Time	11:30 A.M.
Track Condition	Dry

Vehicle weight (LB)	34,500
Front tire pressure (PSI)	50
Rear tire pressure (PSI)	45
Type of fuel used	#2 diesel
Barometric pressure (mm hg.)	751
% Relative humidity	80%
Wind direction	S-SW
Wind velocity	5 MPH
Ambient temperature at track	76°F

Engine speed (RPM)	2800
Output shaft speed (RPM)	488
Ground speed (MPH)	3.5
Reserve tractive force (LB)	8500*
Res. tractive HP (calc.)	79

Temperatures	T(°F)	$\Delta T^{**}$ (°F)
Ambient near vehicle (°F)	72	0
Radiator top tank (°F)	186	114
Radiator bottom tank (°F)	176	104
Engine sump (°F)	225	153
Oil cooler in (°F)	215	143
Oil cooler out (°F)	206	134

\*8500 Lbs was the maximum drawbar pull the vehicle could exert without wheel slippage.

\*\* $\Delta T$  is the difference between the given temperature and the ambient near the vehicle.

FIGURE 2.1

SOUND AT THE OPERATOR'S EAR POSITION

With operator area barrier treatment,  
remote intake, remote exhaust and fan off  
(First Treatment Package)

- High Idle
- △—△ Torque Converter Stall
- High Idle - Lifting 8000 Lb. Load

Octave band or weighted sound level in dB v.s. frequency

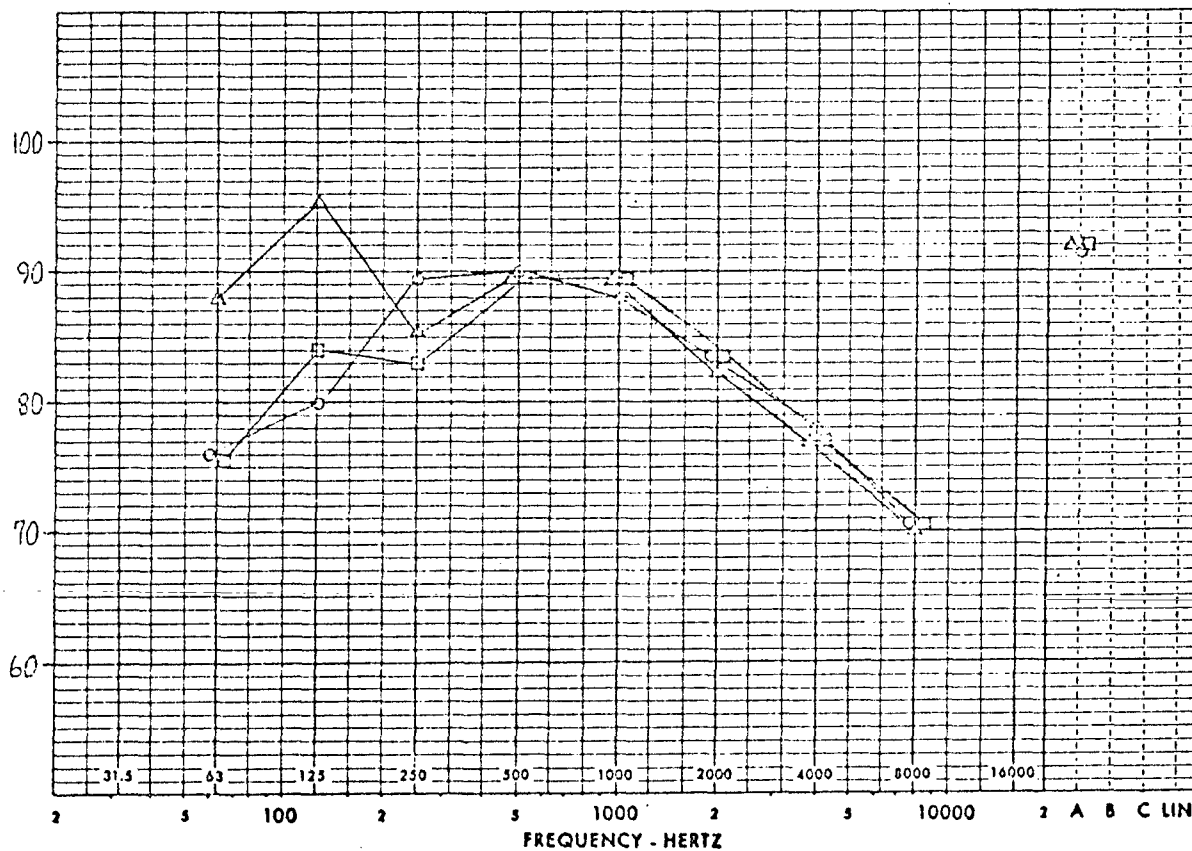


FIGURE 2.2

SOUND AT THE OPERATOR'S EAR POSITION

With operator area barrier treatment,  
stock exhaust, remote intake and fan off

- High Idle
- ▲—▲ Torque Converter Stall
- High Idle - Lifting 8000 lb. Load
- High Idle - First Treatment (from Figure 2.1)

Octave band or weighted sound level in dB v.s. frequency

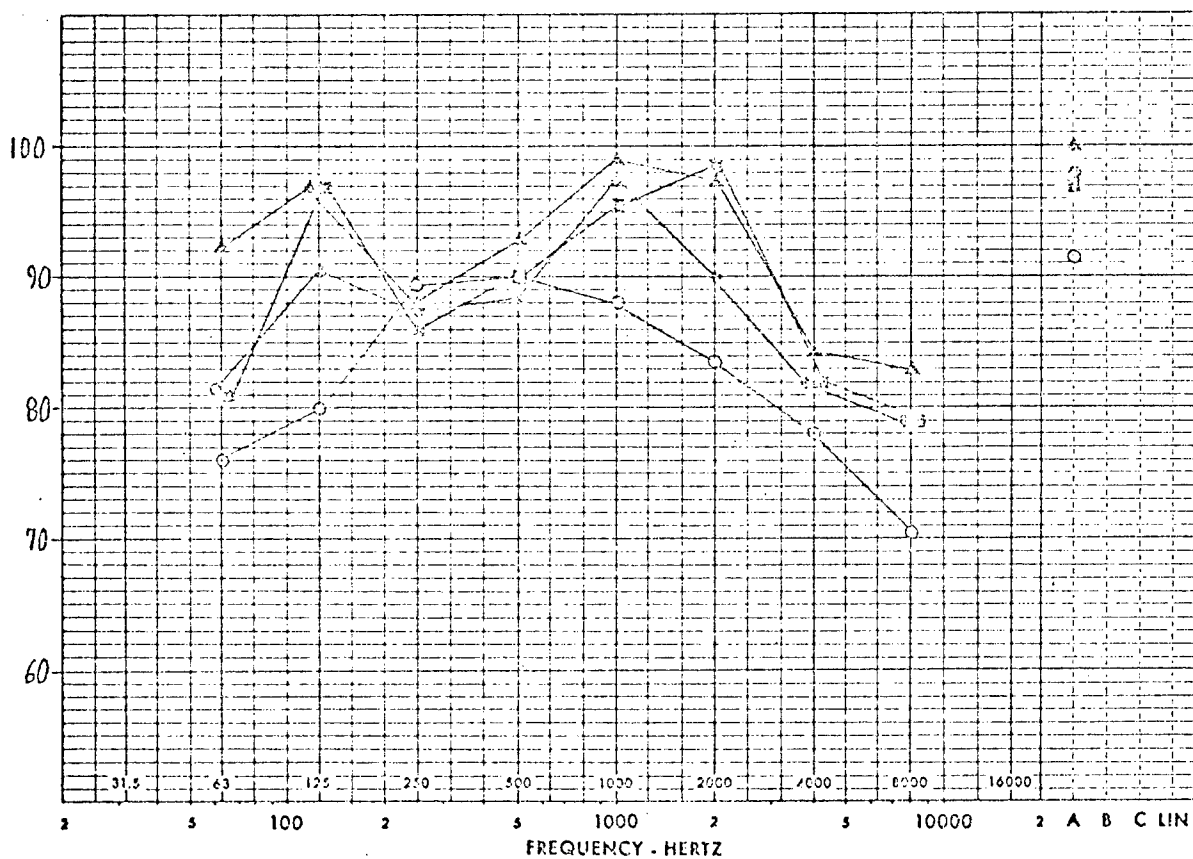


FIGURE 2.3

SOUND AT THE OPERATOR'S EAR POSITION

With no operator area barrier treatment,  
remote intake, remote exhaust and fan off

- High Idle
- ▲—▲ Torque Converter Stall
- High Idle - Lifting 8000 Lb. Load
- High Idle - First Treatment (from Figure 2.1)

Octave band or weighted sound level in dB v.s. frequency

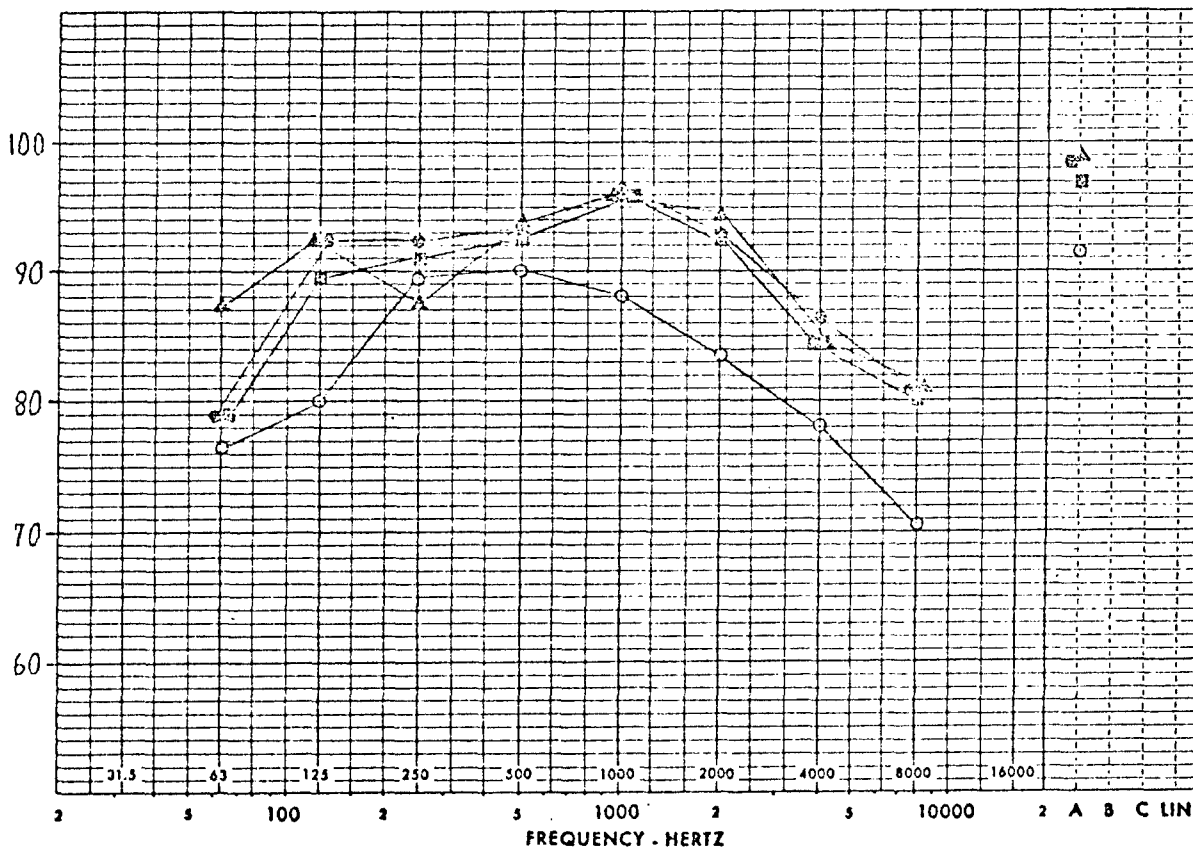


FIGURE 2.4

## SOUND AT THE OPERATOR'S EAR POSITION

With no operator area barrier treatment,  
remote exhaust, remote intake and fan on

- High Idle
- ▲—▲ Torque Converter Stall
- High Idle - Lifting 8000 lb. Load
- High Idle - First Treatment (from Figure 2.1)

Octave band or weighted sound level v.s. frequency

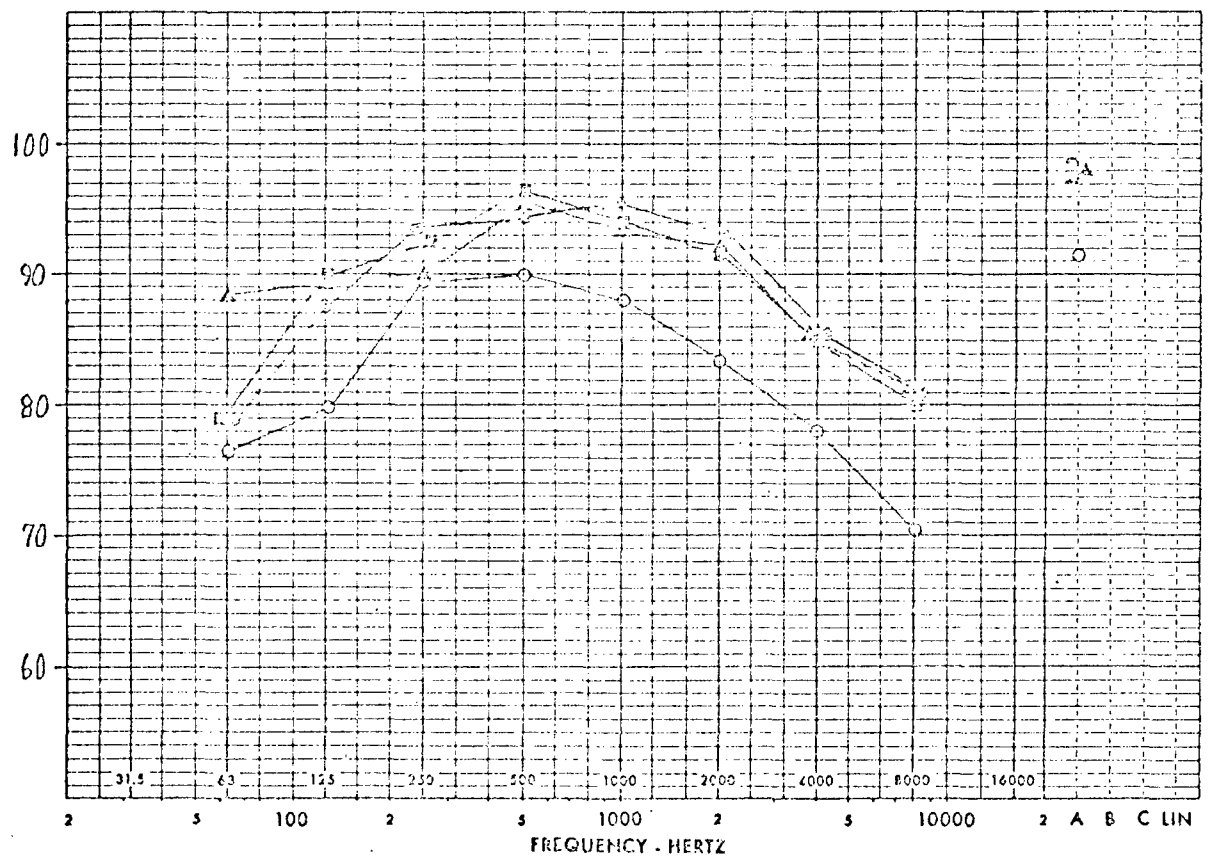


FIGURE 2.5

SOUND AT THE OPERATOR'S EAR POSITION

Second treatment package

- High Idle
- △—△ Torque Converter Stall
- High Idle - Lifting 8000 Lb. Load

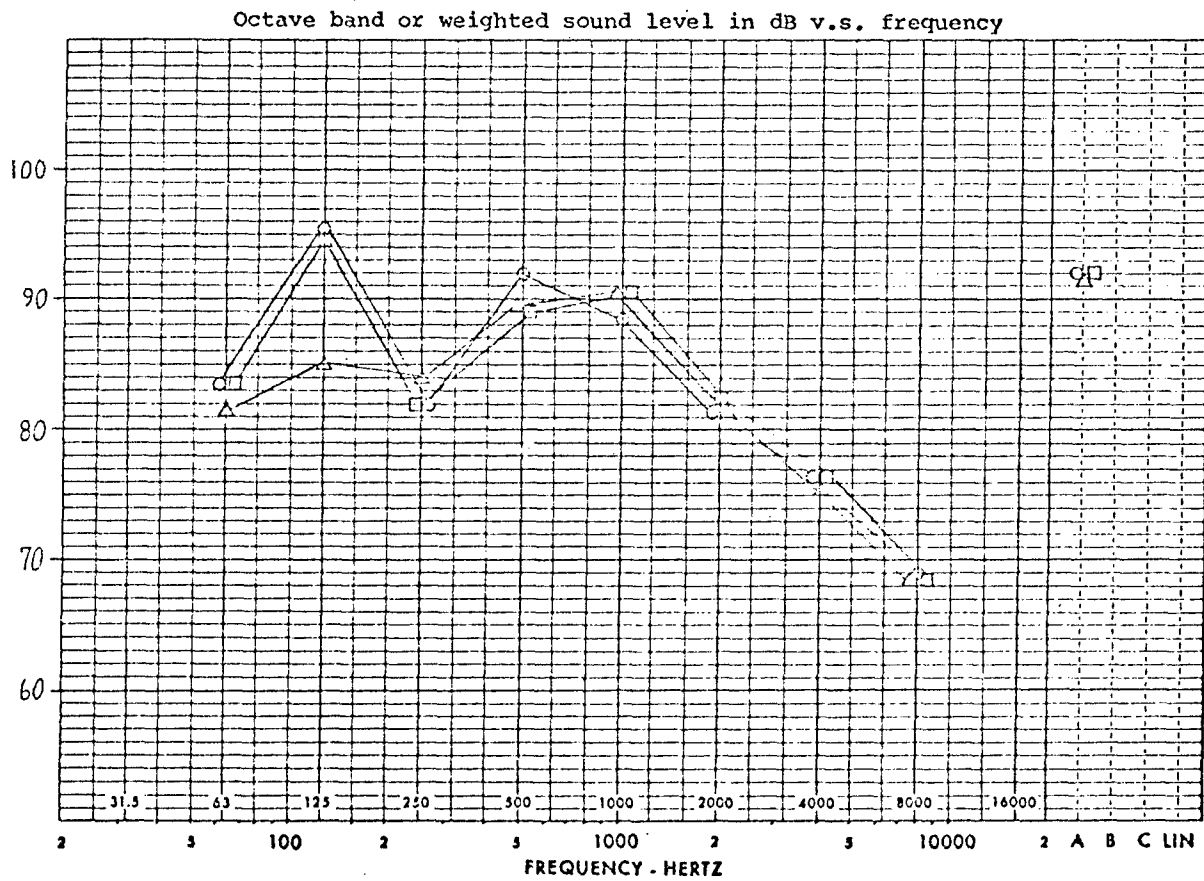




FIGURE 2.6

SOUND IN ENGINE COMPARTMENT

Second treatment package

- High Idle
- High Idle - Propeller Shaft Disconnected
- ◆—◆ High Idle - Truck As Received

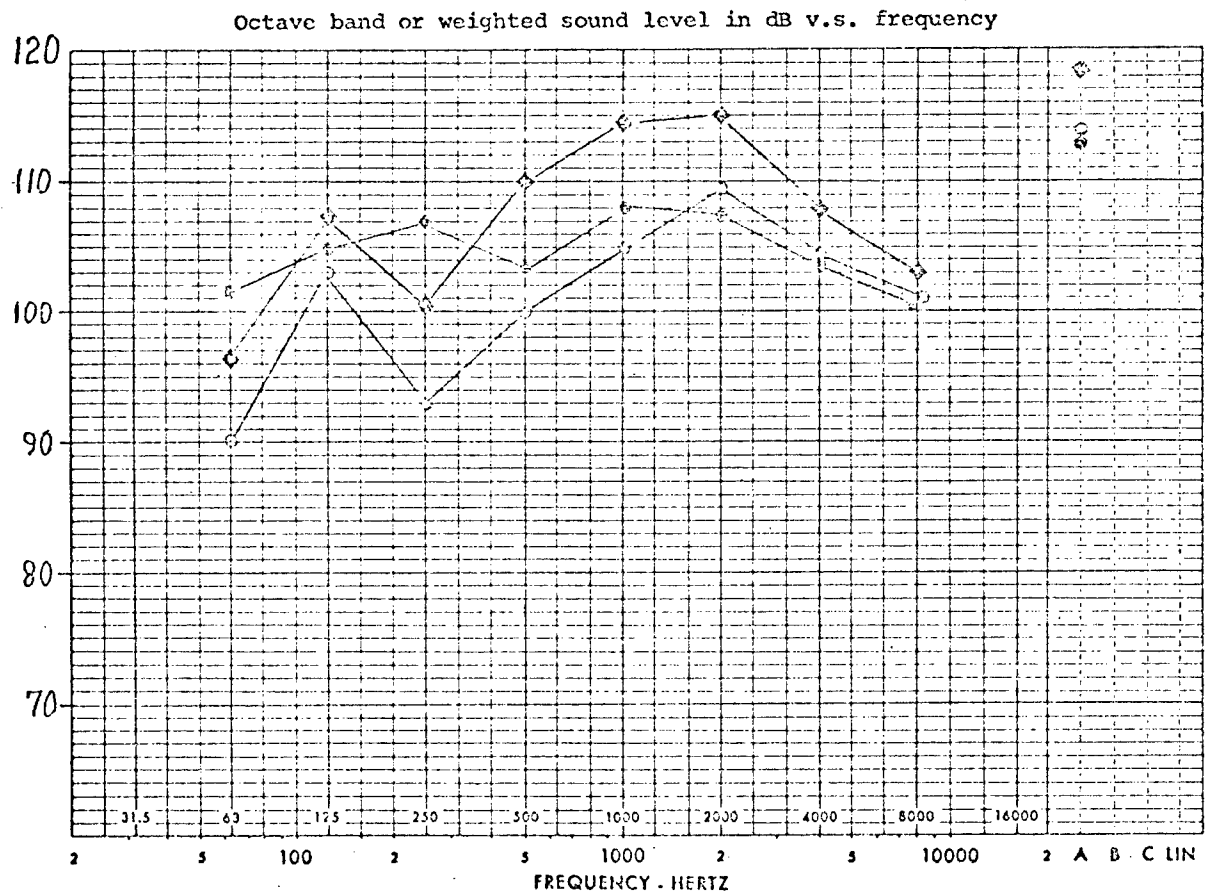


FIGURE 2.7

SOUND AT THE OPERATOR'S EAR POSITION

Second treatment package

- High Idle
- High Idle, Propeller shaft removed

Octave band or weighted sound level in dB v.s. frequency

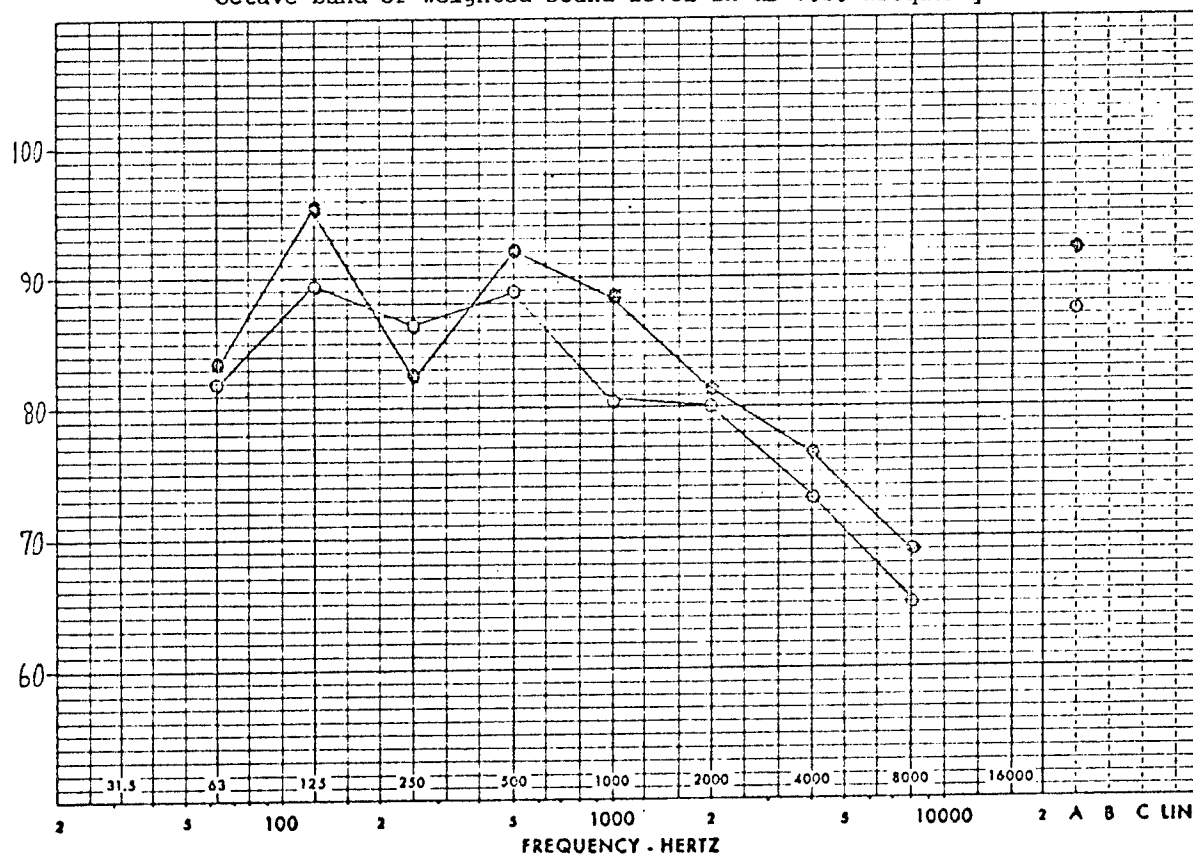


FIGURE 2.8

EFFECT OF ENGINE VIBRATION ON SOUND  
AT THE OPERATOR'S EAR POSITION

Second treatment package with propeller  
shaft from engine to transmission removed

- High idle - hard engine mounting
- High idle - isolated engine

Octave band or weighted sound level in dB v.s. frequency

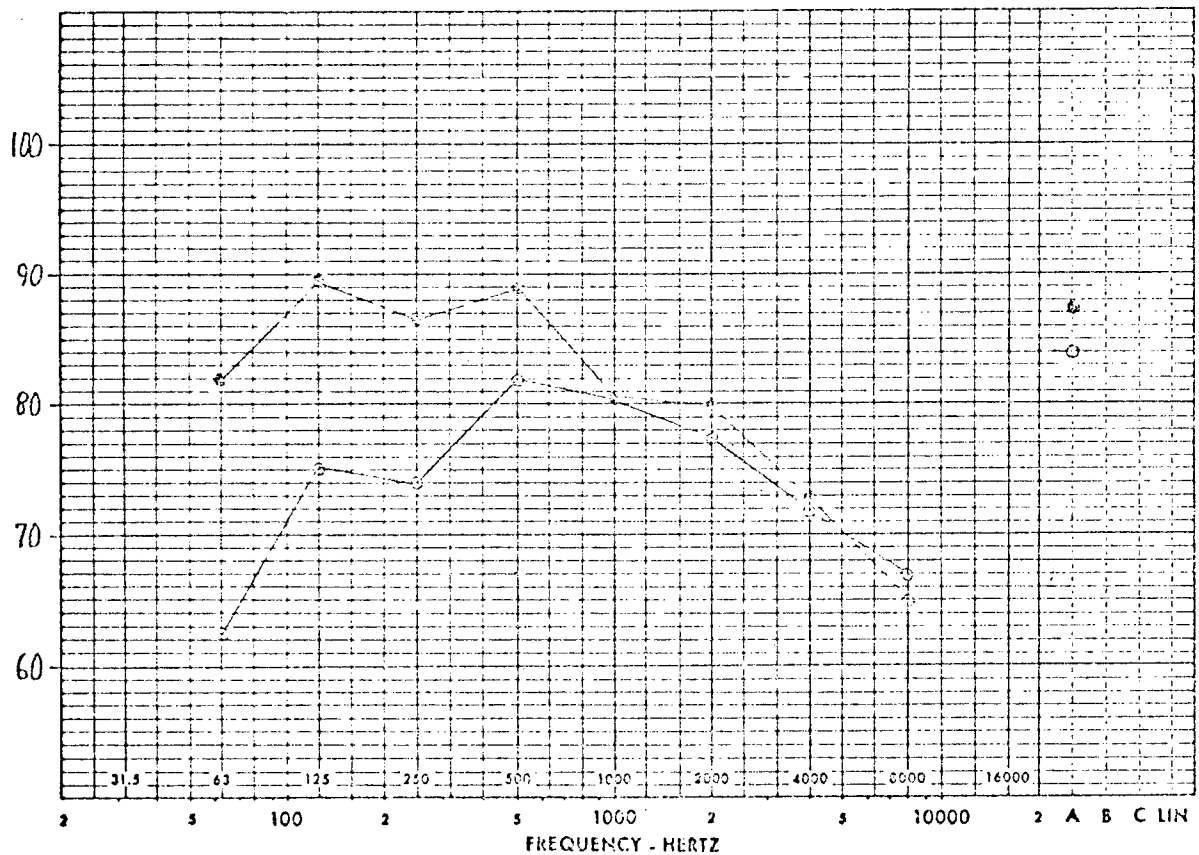


FIGURE 2.9

EFFECT OF TRANSMISSION VIBRATION ON  
SOUND AT THE OPERATOR'S EAR POSITION

Third treatment package

- — High idle - propeller shaft from engine to transmission removed
- — High idle - propeller shaft in place with hard mounted transmission

Octave band or weighted sound level in dB v.s. frequency

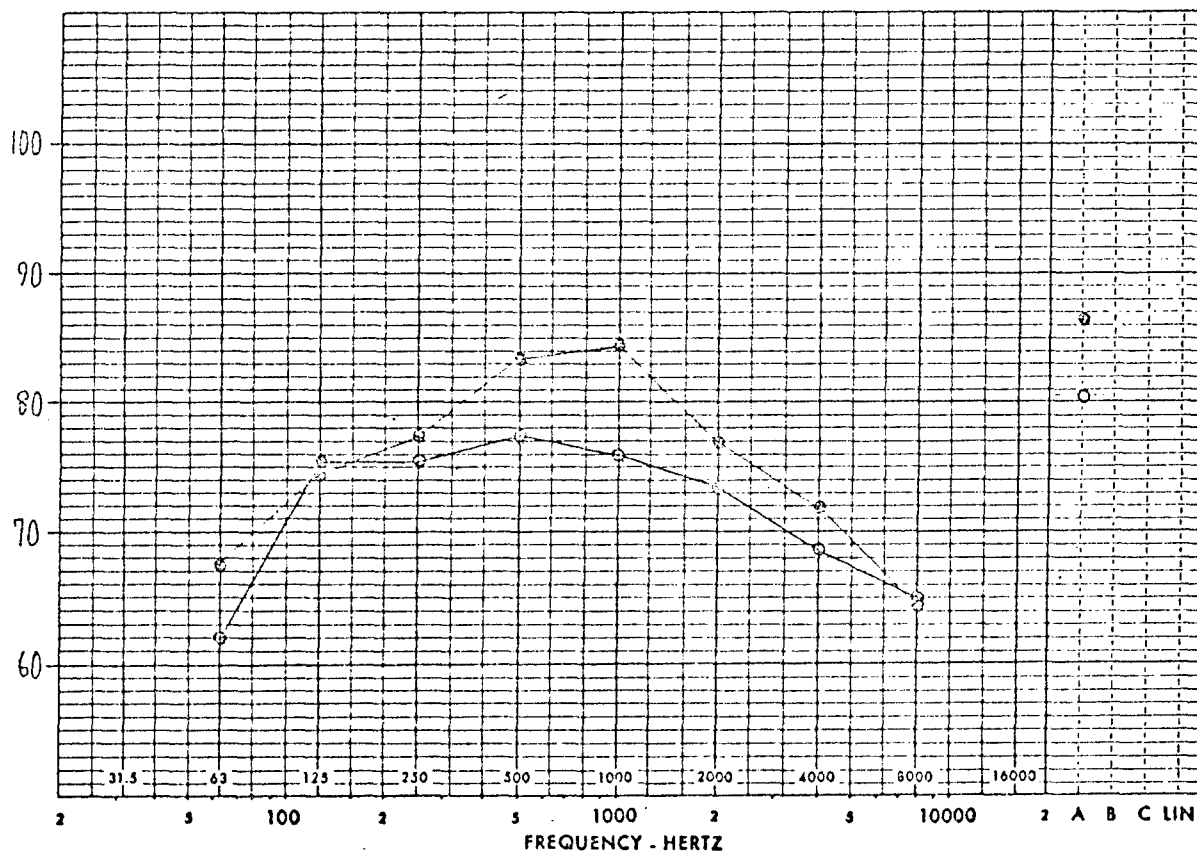


FIGURE 2.10

## EFFECT OF INTAKE ON SOUND AT THE OPERATOR'S EAR POSITION

Third Treatment package with propeller shaft  
from engine to transmission removed

- — ○ High Idle - remote intake
- — ■ High Idle - shell radiation from air cleaner only
- — ● High Idle - completely stock intake

Octave band or weighted sound level in dB v.s. frequency

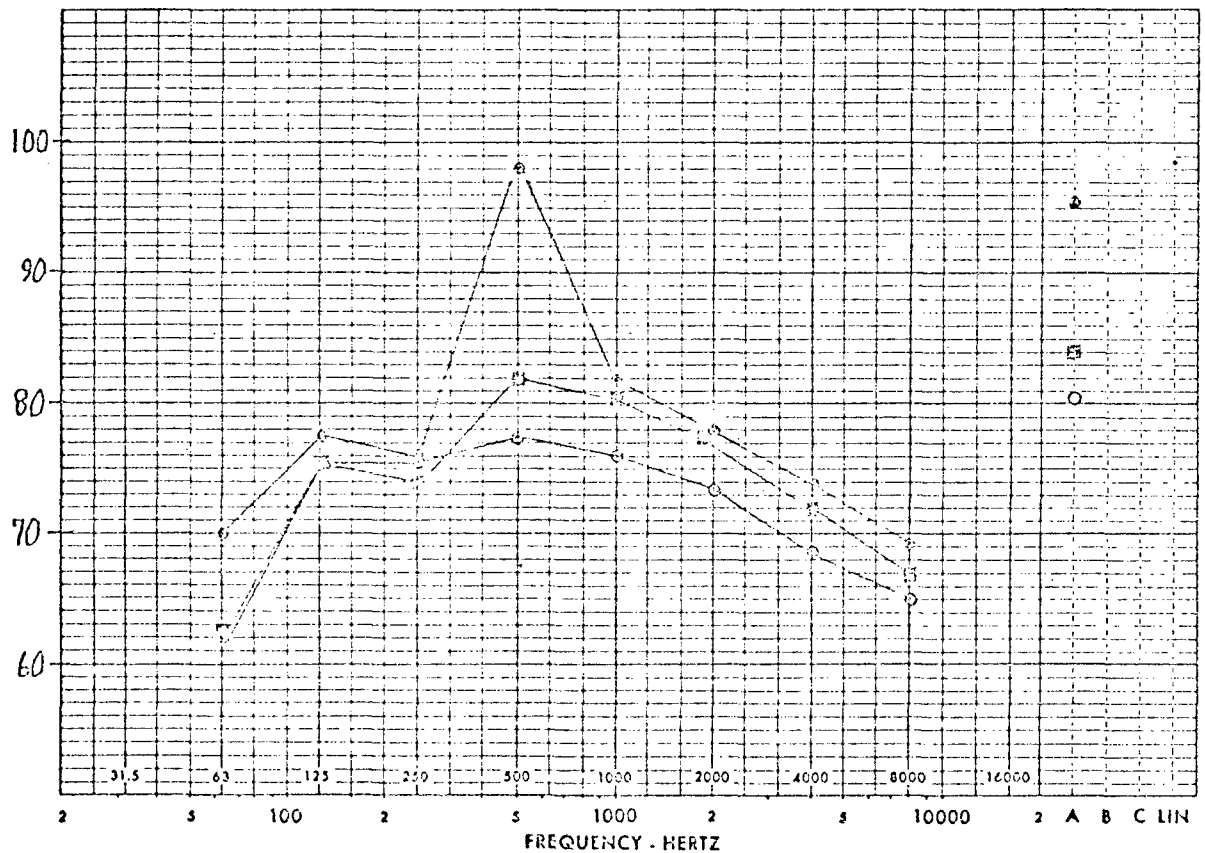


FIGURE 2.11

EFFECT OF RADIATOR OPENING ON  
SOUND AT THE OPERATOR'S EAR POSITION

Third treatment package with propeller shaft  
from engine to transmission removed

- High idle - with large treated radiator baffle
- High idle - no radiator baffle

Octave band or weighted sound level in dB v.s. frequency

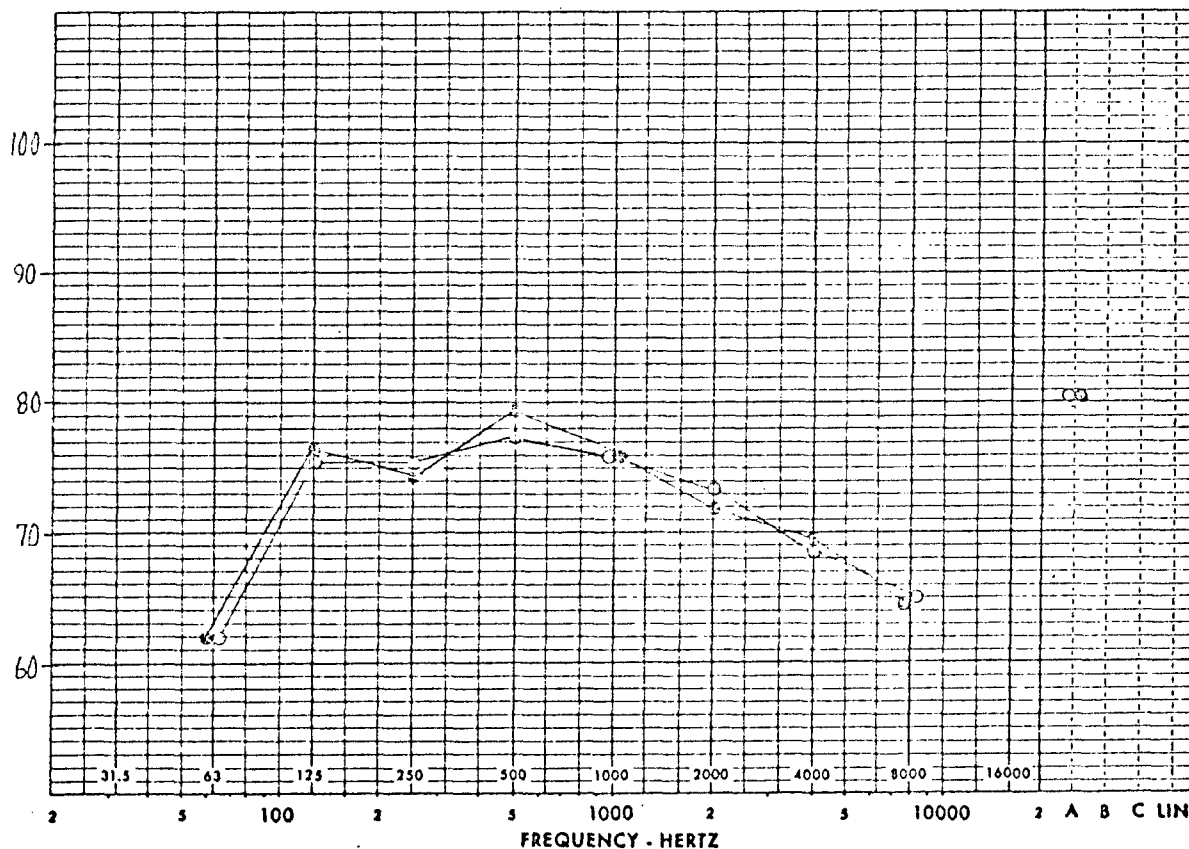


FIGURE 2.12

EFFECT OF ENGINE FAN ON SOUND  
AT THE OPERATOR'S EAR POSITION

Third treatment package with no radiator baffle  
and propeller shaft to transmission removed

- — ○ High idle - fan off
- — ● High idle - fan on

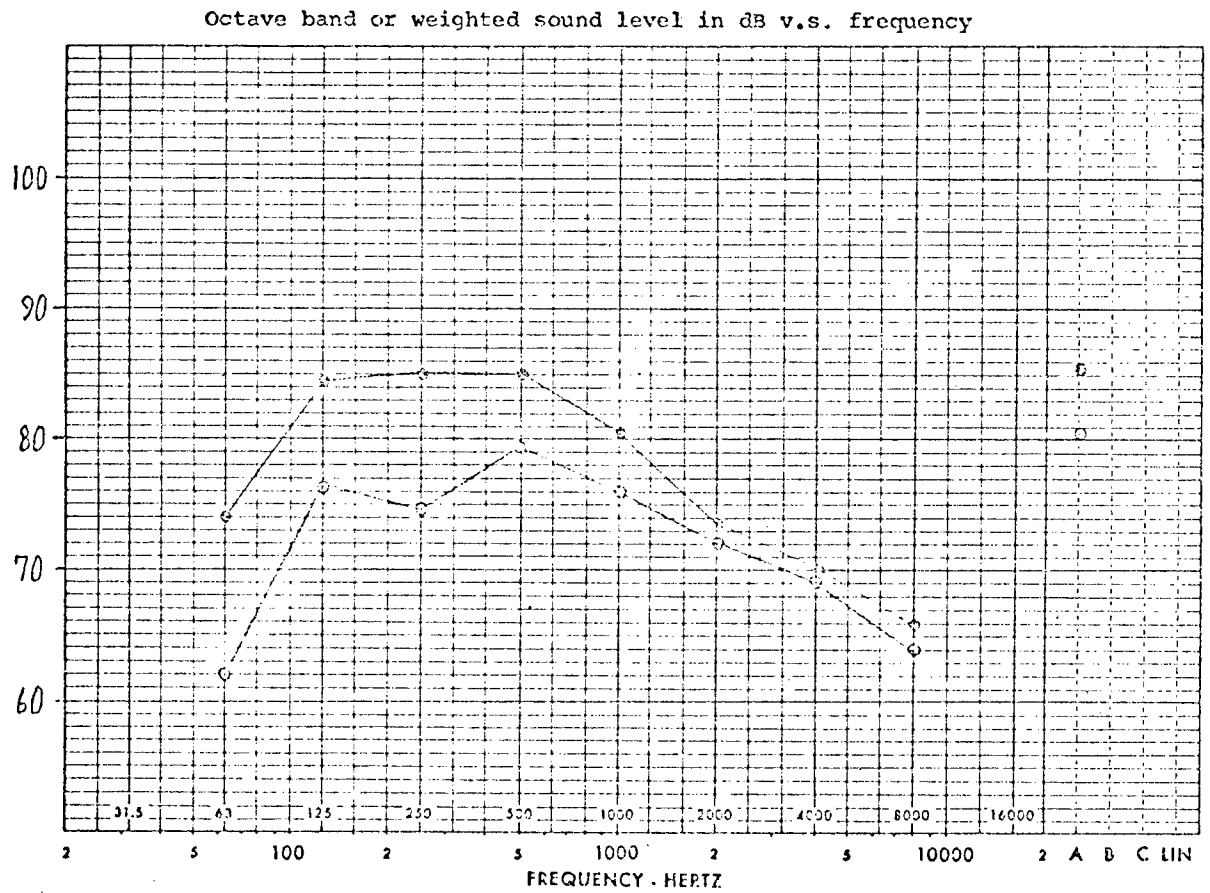


FIGURE 2.13

EFFECT OF OPEN UNDERSIDE ON SOUND  
AT THE OPERATOR'S EAR POSITION

Third treatment package with propeller shaft on

- High idle - fan off, underside sealed
- High idle - fan on, underside sealed
- High idle - fan off, underside open
- High idle - fan on, underside open

Octave band or weighted sound level in dB v.s. frequency

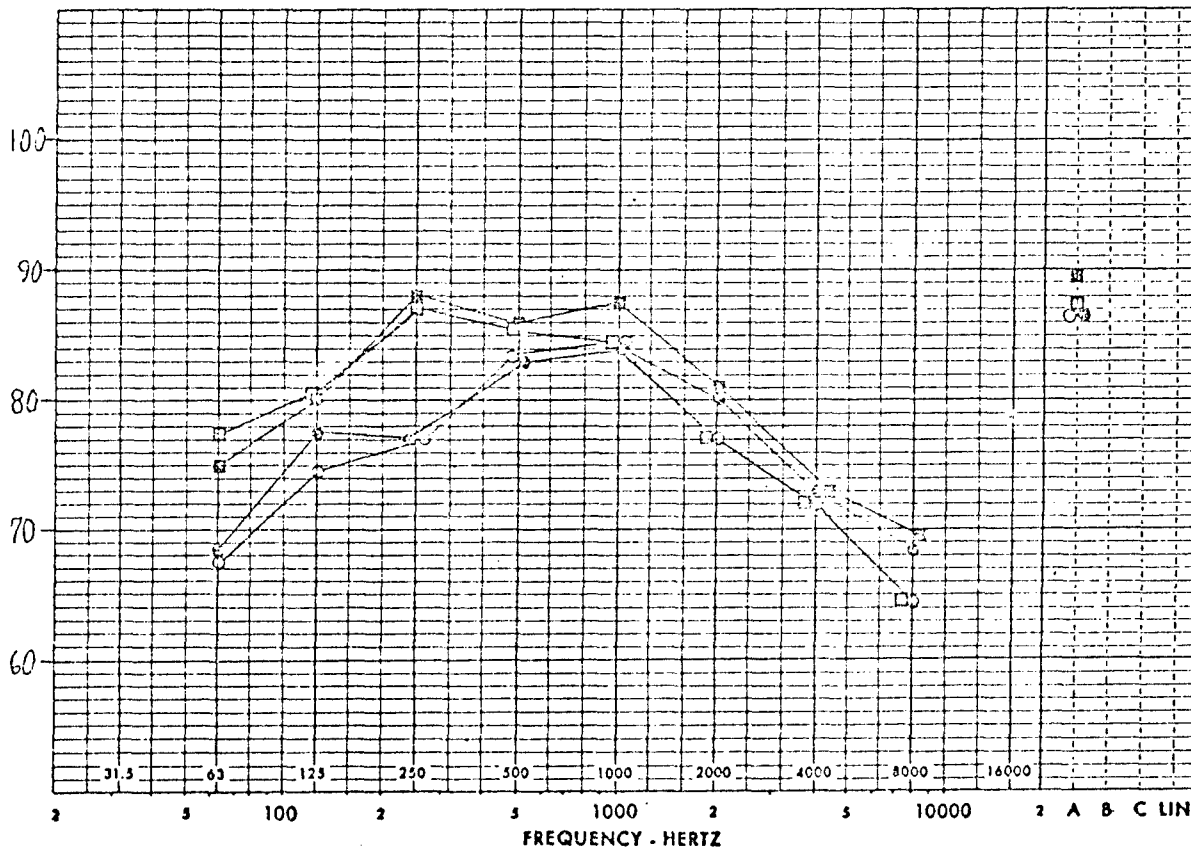




FIGURE 2.14

EFFECT OF VIBRATION OF NEARBY PANELS ON  
SOUND AT THE OPERATOR'S EAR POSITION

- (1) —○— Fourth treatment package - high idle.
- (2) —△— (1) with barrier material removed from hydraulic resevoir.
- (3) —◇— (1) with barrier material removed from all nearby panels.
- (4) —■— (3) with holes in panels open.

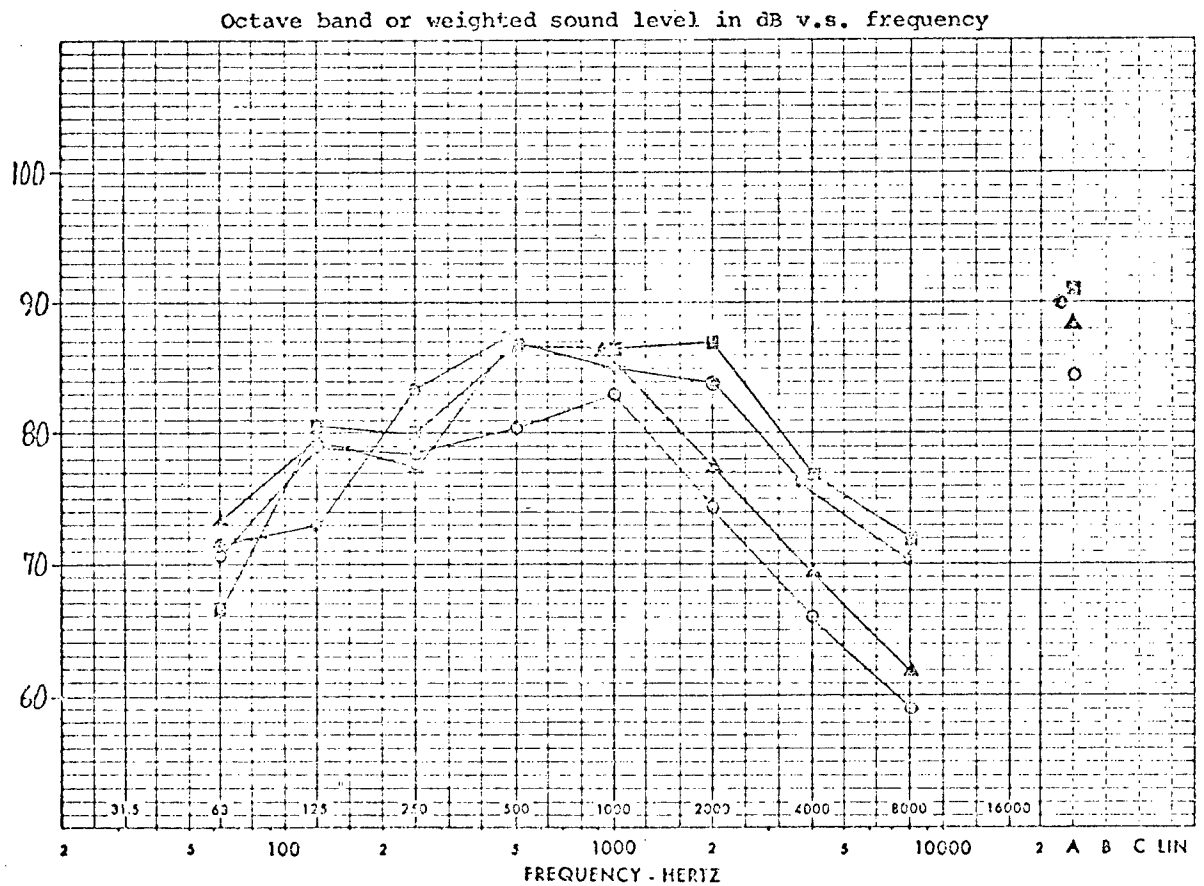


FIGURE 2.15

EFFECT OF BOOM AREA OPENING ON SOUND  
AT THE OPERATOR'S EAR POSITION

- (1) —○— Fourth treatment package - high idle
- (2) —□— (1) with lower boom area open
- (3) —●— (1) with entire front boom area open

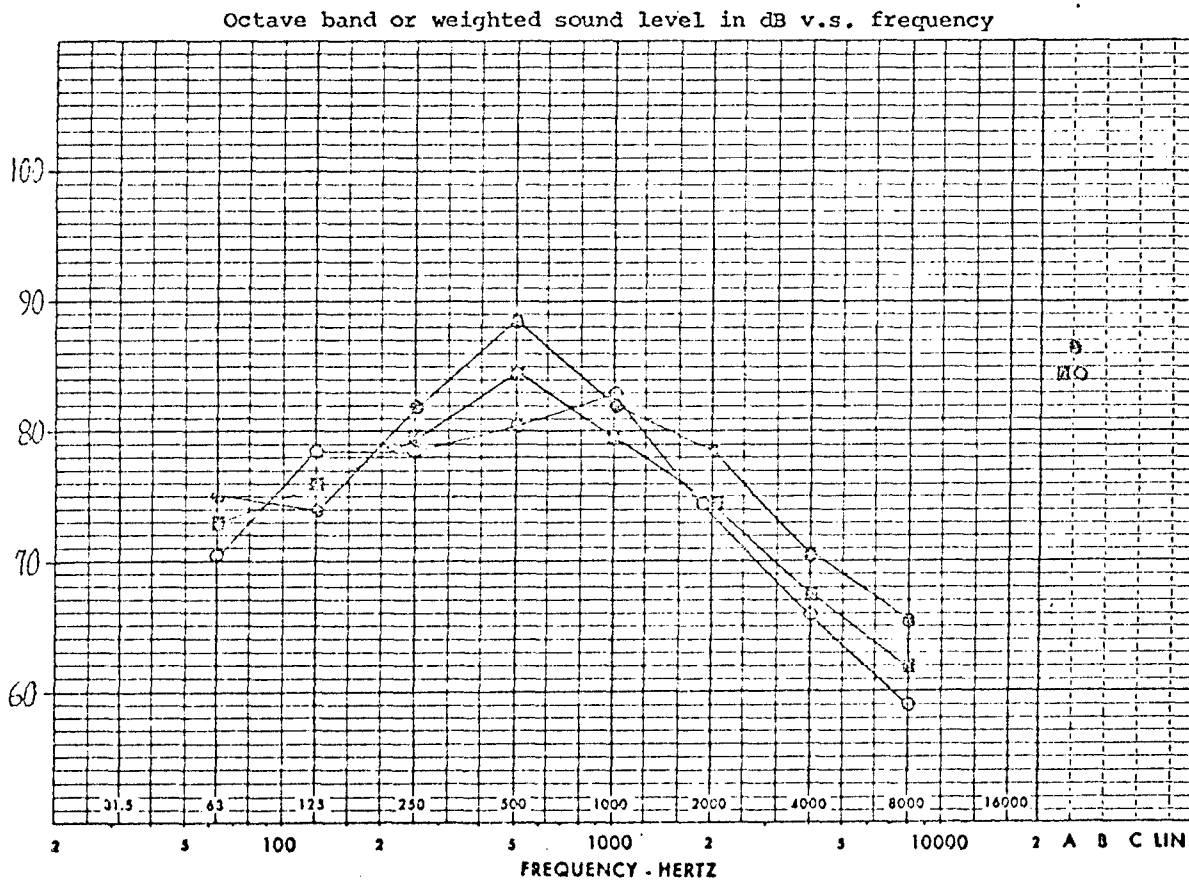


FIGURE 2.16

EFFECT OF OPEN UNDERSIDE ON SOUND  
AT THE OPERATOR'S EAR POSITION

- (1) —○— Fourth treatment package - high idle
- (2) —□— (1) with rear underside open
- (3) —◇— (1) with entire underside open

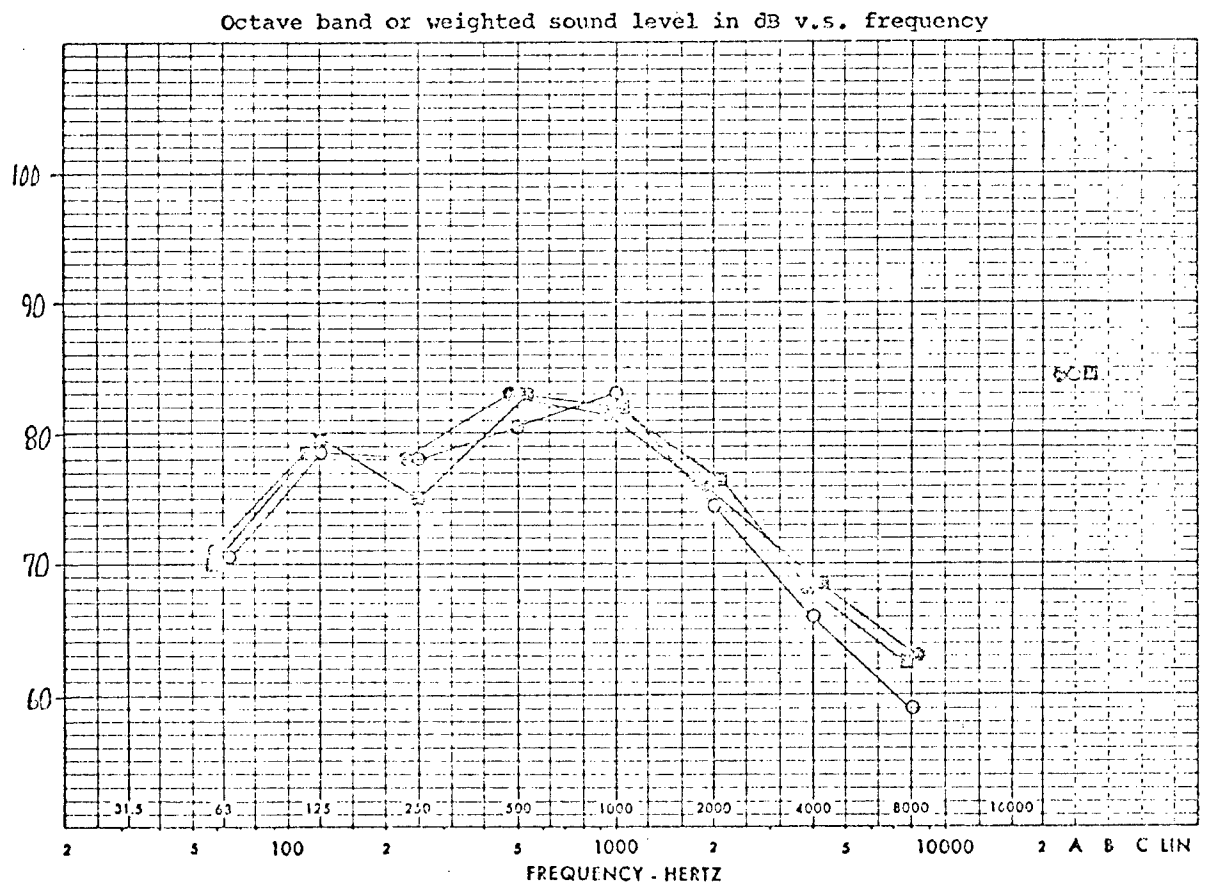


FIGURE 2.17

EFFECT OF ENGINE SIDE PANEL CRACKS  
ON SOUND AT THE OPERATOR'S EAR POSITION

- (1) —○— Fourth treatment package - high idle  
(2) —●— (1) with engine side panel cracks unsealed

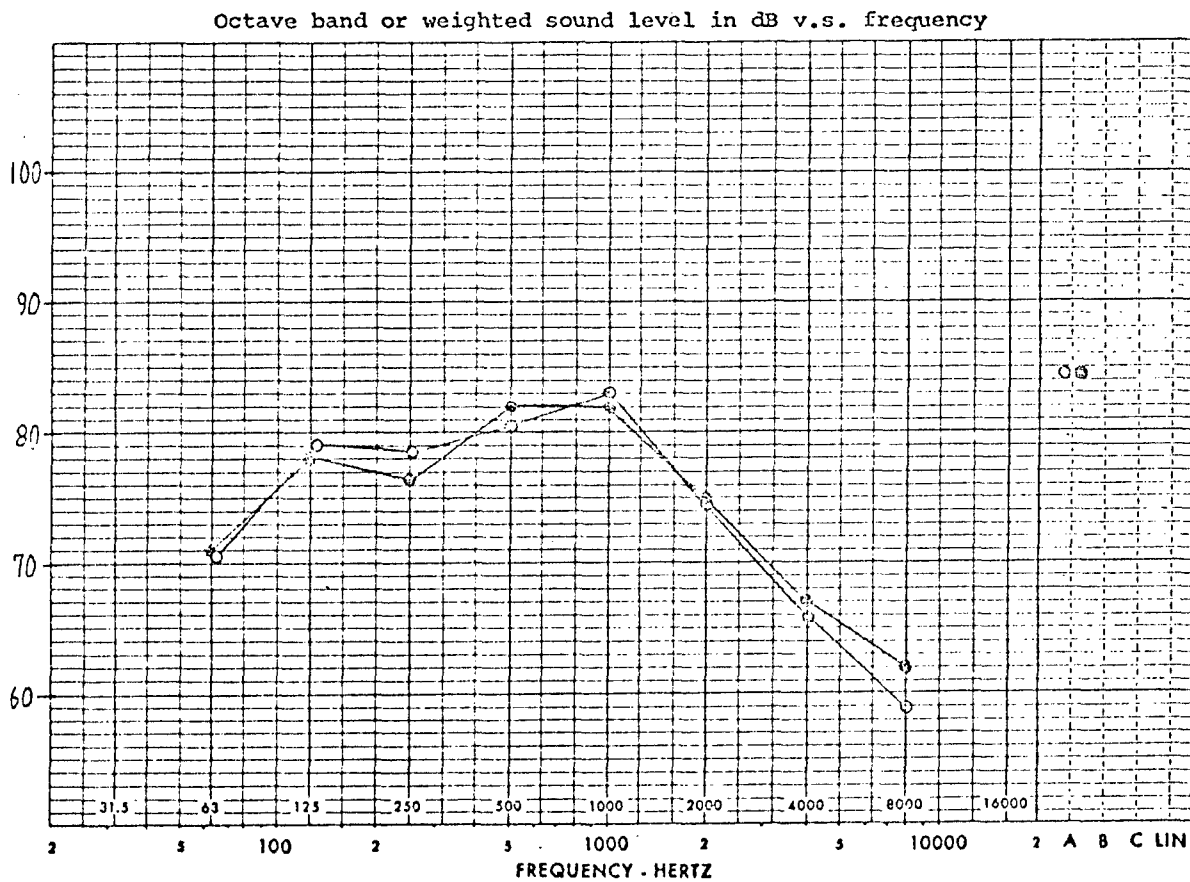


FIGURE 2.18

EFFECT OF HYDRAULIC SYSTEM AND TORQUE  
CONVERTER ON SOUND AT THE OPERATOR'S EAR POSITION

- (1) —○— High idle - fifth treatment package  
 (2) —□— (1) with hydraulic system removed  
 (3) —○— (2) with prop shaft off (Transmission disabled)  
 (4) —▽— Case (3) from Figure 2.14.

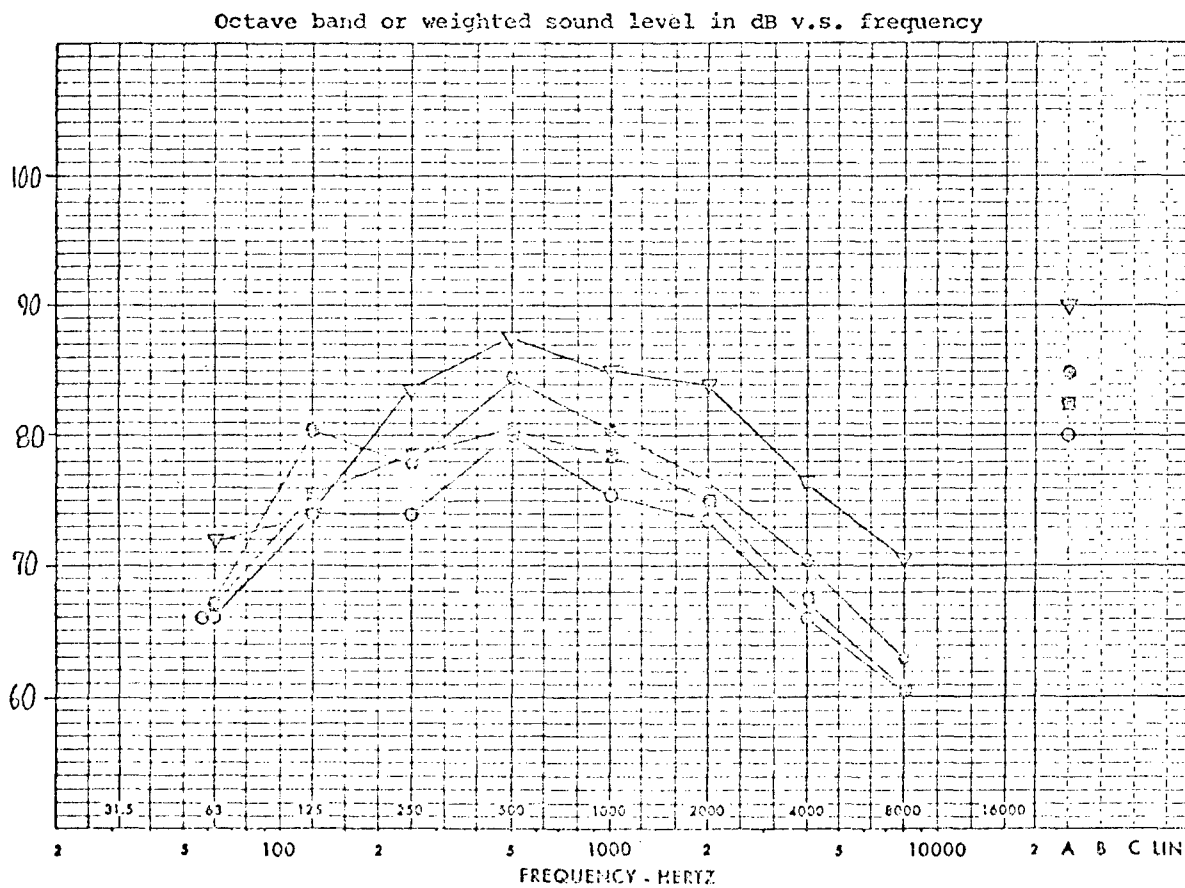


FIGURE 2.19

SOUND IN ENGINE COMPARTMENT  
(8.5" BEHIND HYDRAULIC STEERING PUMP)

- (1) —○— High idle - fifth treatment package  
(2) —□— (1) with hydraulic system removed  
(3) —○— (2) with prop shaft off (T.C. and gears stopped)

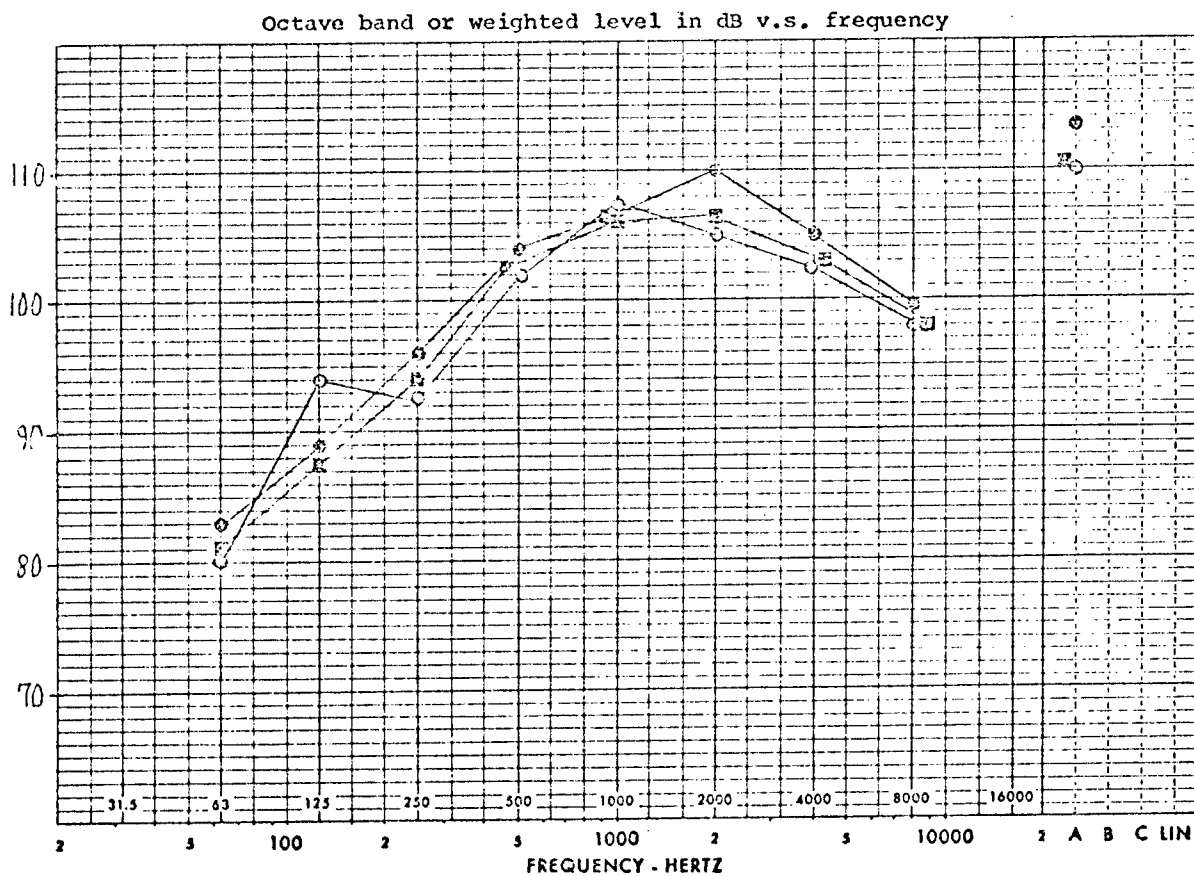


FIGURE 2.20

SOUND 6" BELOW THE MAIN HYDRAULIC PUMP  
AND HYDRAULIC BRAKE PUMP

- (1) —●— High idle - fifth treatment package
- (2) —■— (1) with hydraulic system removed
- (3) —○— (2) with prop shaft off (T.C. and gears stopped)

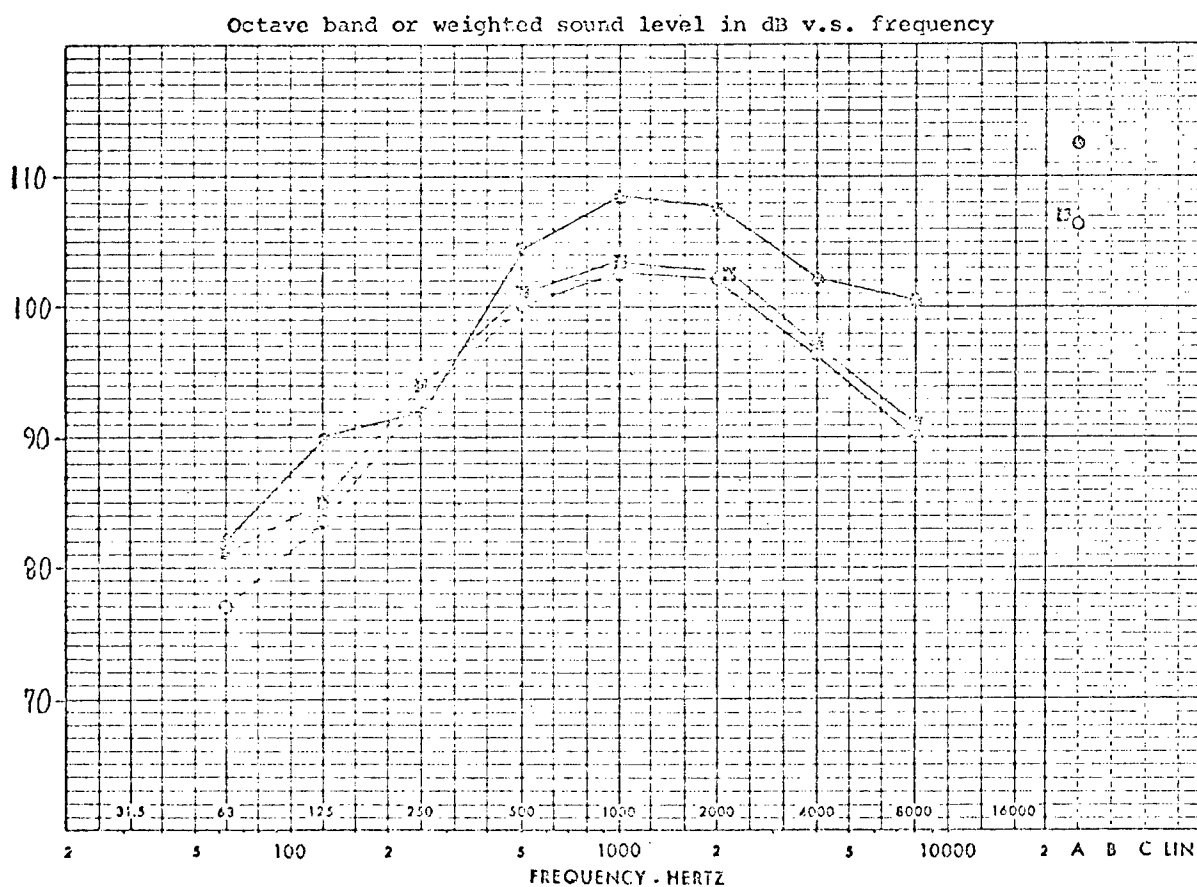


FIGURE 2.21

SOUND AT THE OPERATOR'S EAR POSITION

With Stemco mufflers, relocated air intake, damped hood and lift control lever panel, isolated engine and transmission and all holes nearest operator closed.  
Fan on.

- (1) —○— High Idle
- (2) —△— Torque Converter Stall
- (3) —□— High Idle - Lifting 8000 Lb. Load
- (4) —●— High Idle As Received

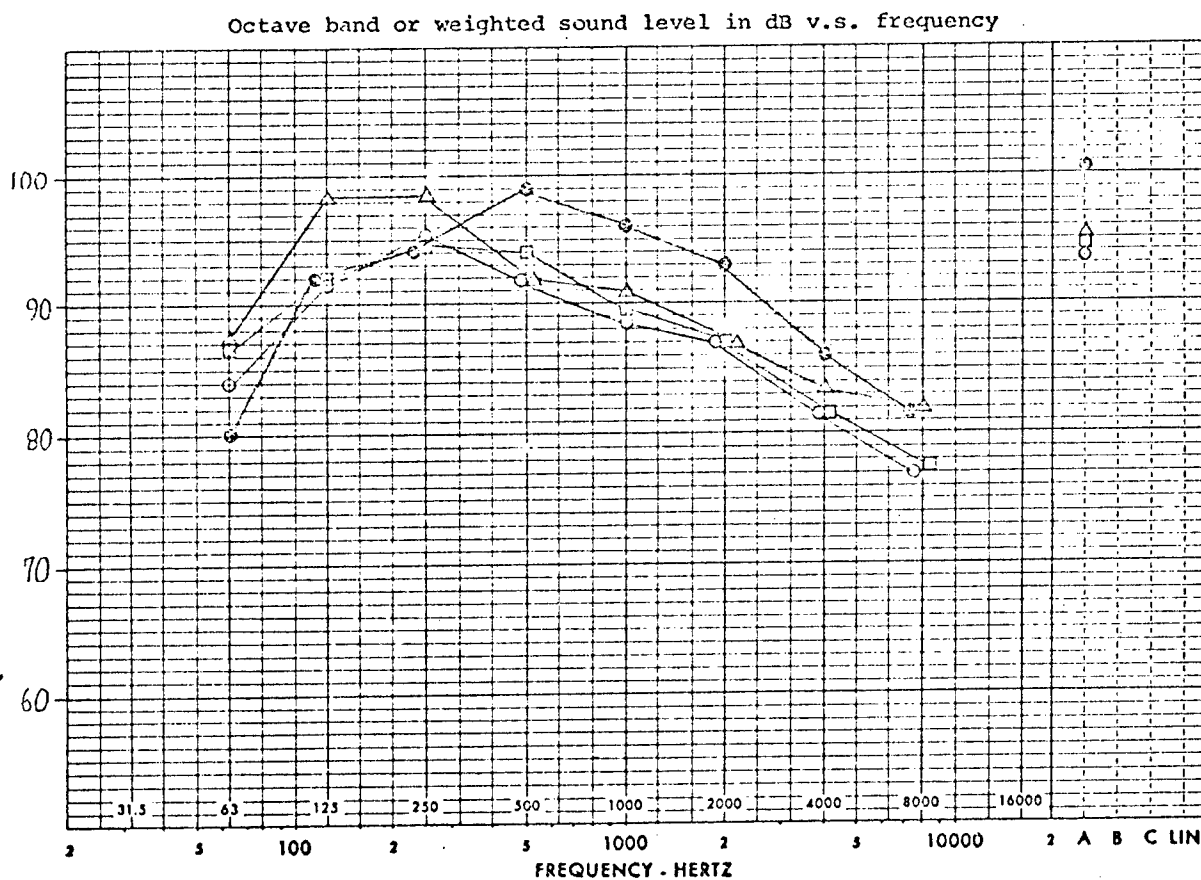




FIGURE 2.22

SOUND AT THE OPERATOR'S EAR POSITION

With Stemco mufflers, relocated air intake, damped hood and lift control lever panel, isolated engine and transmission and largest holes nearest operator closed. (Lift control lever slots and right side panel access holes closed) Fan off.

- (1) —□— High Idle
- (2) —△— (1) plus flat barriers on upper boom
- (3) —○— (2) plus engine compartment absorptive treatment.

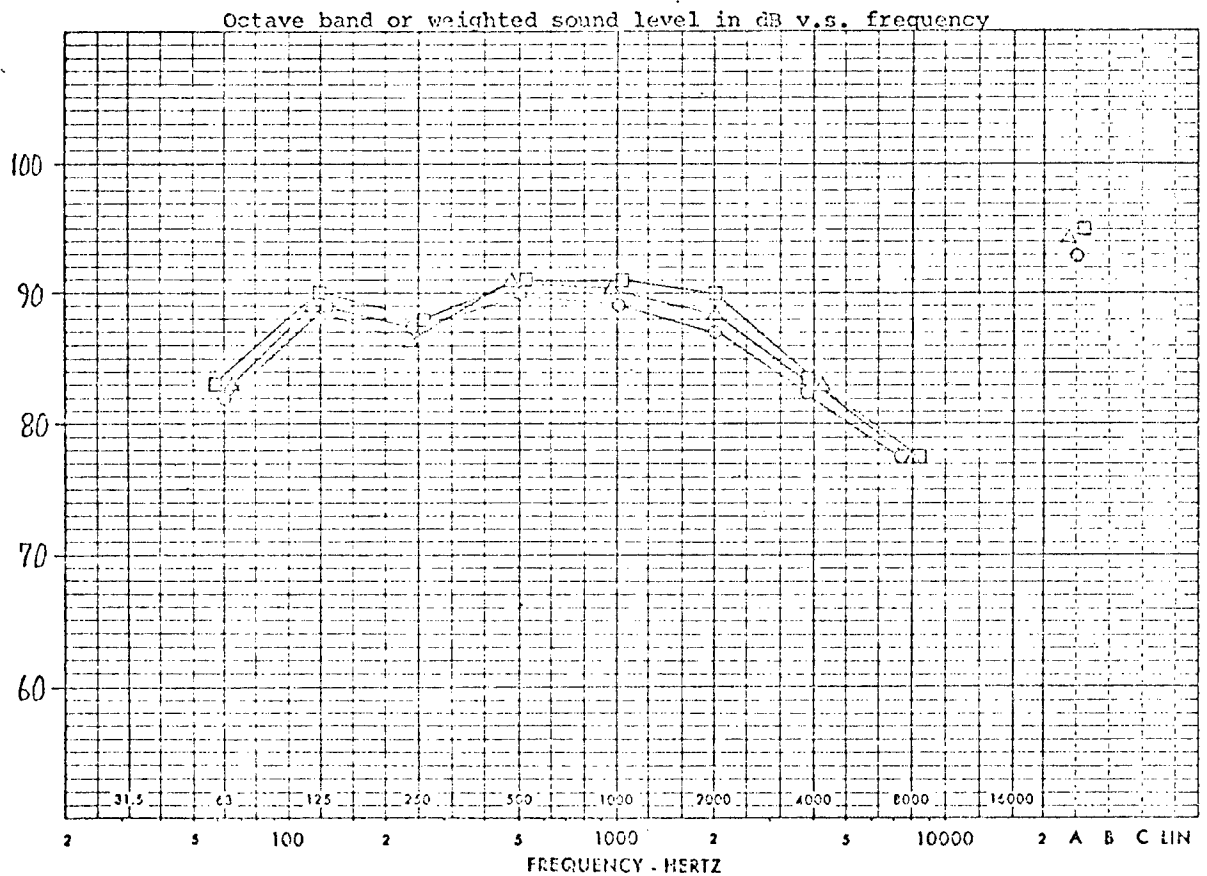


FIGURE 2.23 TABLE OF TREATMENTS

x = Treatment employed for all tests on graph (s)  
 (x) = Treatment removed for some tests on graph (s)

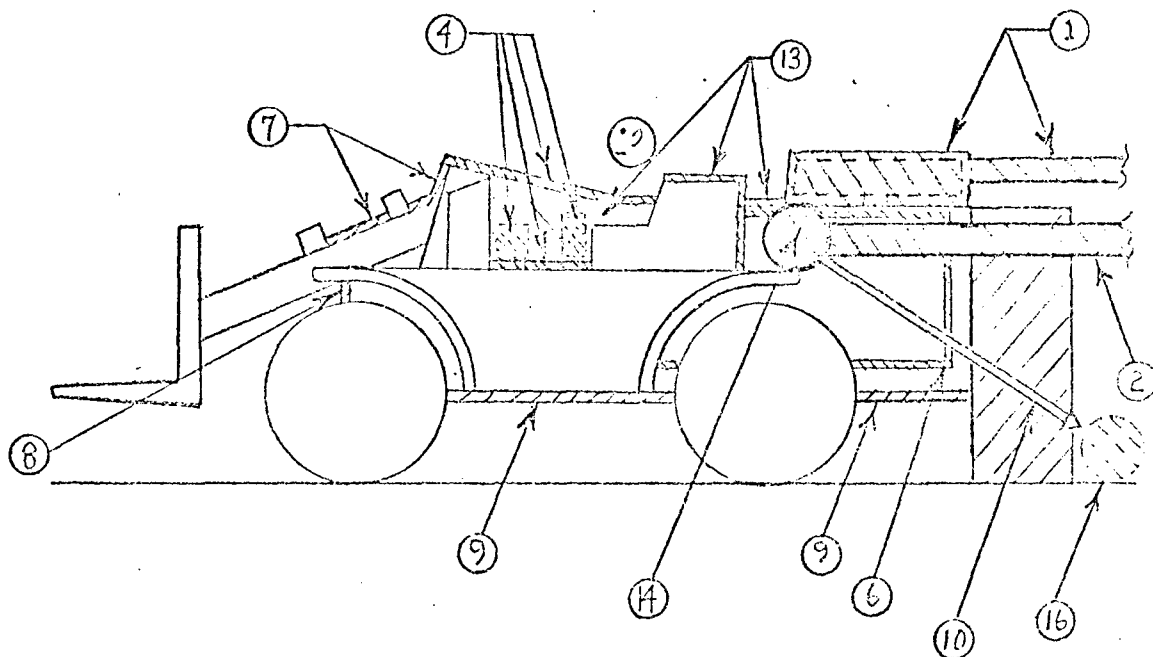
FIGURE NO. (S)	2.1	2.2	2.3	2.4	2.5 2.6	2.7	2.8	2.9	2.10	2.11	2.12	2.13	2.14 2.15 2.16 2.17 2.18 2.19 2.20
TREATMENT PACKAGE NO.	1	1	1	1	2	2	2	3	3	3	3	4	5
Element of Treatment													
1. Remote and wrapped exhaust	x		x	x	x	x	x	x	x	x	x	x	x
2. Remote air intake	x	x	x	x	x	x	x	(x)	(x)	x	x	x	x
3. Fan off	x	x	x		x	x	x	x	x	(x)	x	x	x
4. Nearby holes closed	x	x			x	x	x	x	x	x	x	(x)	x
5. Absorption in engine comp.					x	x	x	x	x	x	x	x	x
6. Sealed top and sides of eng. comp.					x	x	x	x	x	x	x	(x)	
7. Sealed upper boom area					x	x	x	x	x	x	x	(x)	x
8. Sealed lower boom area					x	x	x	x	x	x	x	(x)	
9. Sealed underside					x	x	x	x	x	x	(x)	(x)	
10. Radiator baffle					x	x	x	x	x	(x)		x	x
11. Isolated engine						x	x	x	x	x	x	x	x
12. Isolated transmission												x	x
13. Barrier mat'l on nearby panels								x	x	x	x	(x)	x
14. Wrapped air intake canister								x	(x)	x	x	x	x
15. Propeller shaft removed									x	x	x		(x)
16. Air cleaner on floor of room												x	
17. Intake on right side of vehicle													x
18. Hydraulic systems removed													(x)

FIGURE 2.24

EXTERNALLY VISIBLE TEMPORARY NOISE CONTROLS  
USED IN THE FIVE TREATMENT PACKAGES OF PHASE II

See also Figure 2.23.

View is of left side (operator side) of vehicle.

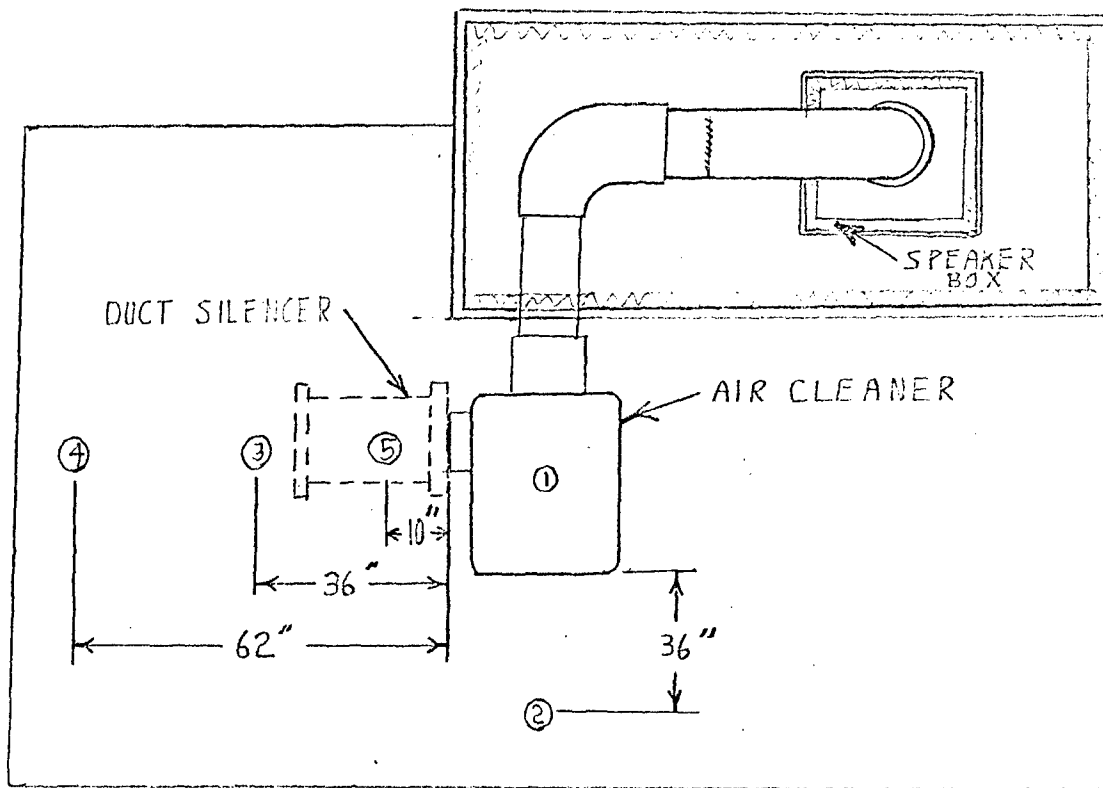


Element of Treatment

- 1 Remote exhaust
- 2 Remote air intake
- 3 Fan off (not shown)
- 4 Nearby holes closed
- 5 Absorption in engine comp. (not shown)
- 6 Sealed top and sides of engine comp.
- 7 Sealed upper boom area
- 8 Sealed lower boom area
- 9 Sealed underside

Element of Treatment

- 10 Radiator baffle (lined labyrinth)
- 11 Isolated engine (not shown)
- 12 Isolated transmission (not shown)
- 13 Barrier mat'l on nearby panels
- 14 Wrapped air cleaner canister
- 15 Propeller shaft removed (not shown)
- 16 Air cleaner on floor of room
- 17 Intake on right side of vehicle (not shown)
- 18 Hydraulic systems removed (not shown)



VIEW A-A (PLAN VIEW)

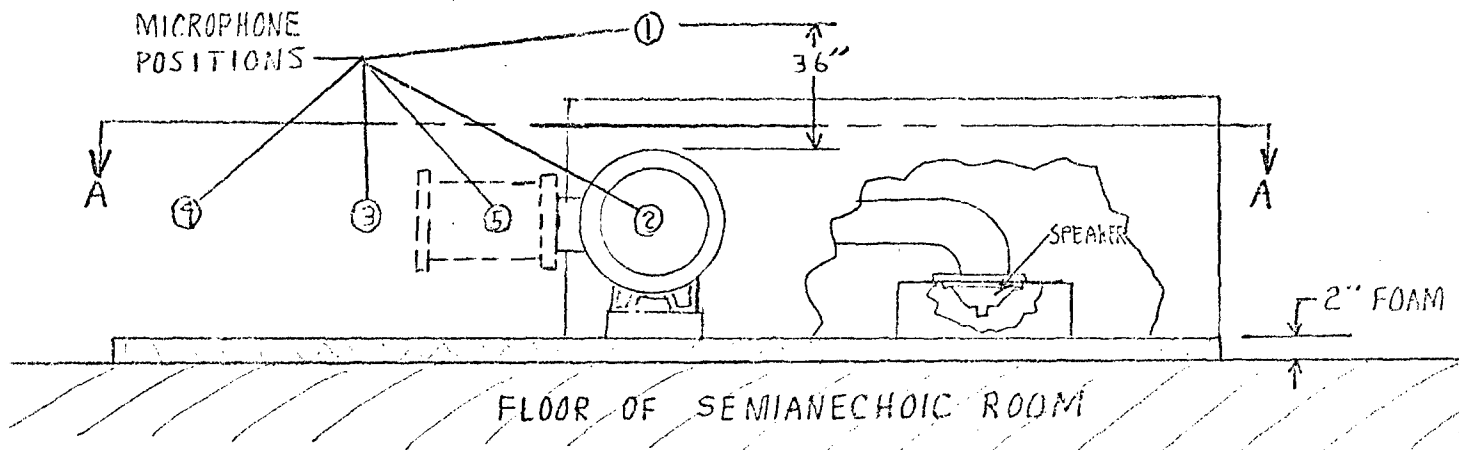


FIGURE 3.1.1

# ORIGINAL AIR CLEANER AND SILENCER TEST SET-UP

The purpose of the two inch foam was to reduce unwanted reflection of sound from the floor. Tests confirmed that the inverse square law (-6 dB per double distance) was being approximately followed above the foam so it was considered adequate.

FIGURE 3.1.1.2  
 INSERTION LOSS OF TWO AIR CLEANERS

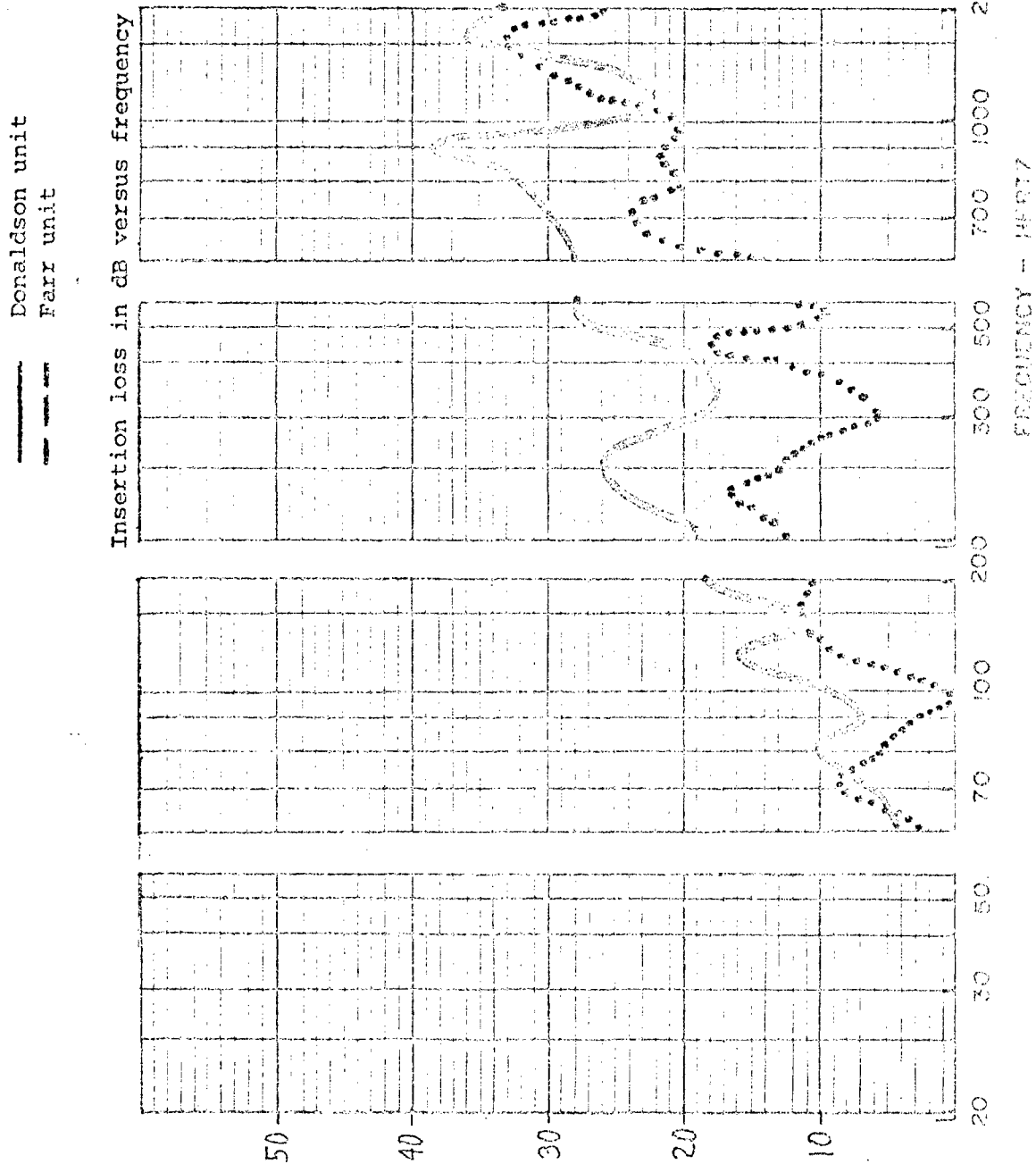


FIGURE 3.1.3

INSERTION LOSS OF DONALDSON AIR CLEANER  
PLUS UNIVERSAL SILENCER

— Air cleaner and silencer

..... Air cleaner only (from Figure 3.1.2)

Insertion loss in dB versus frequency

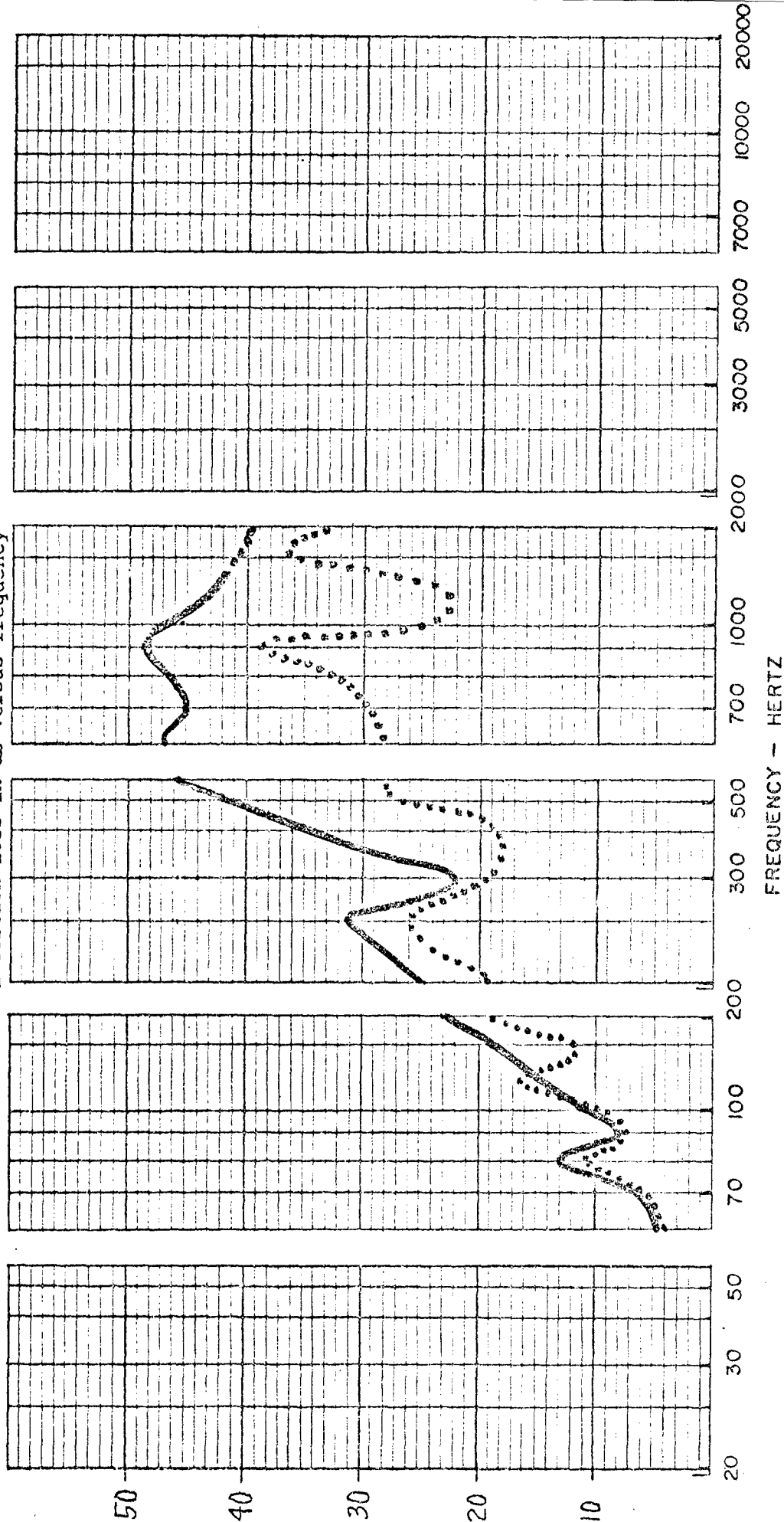
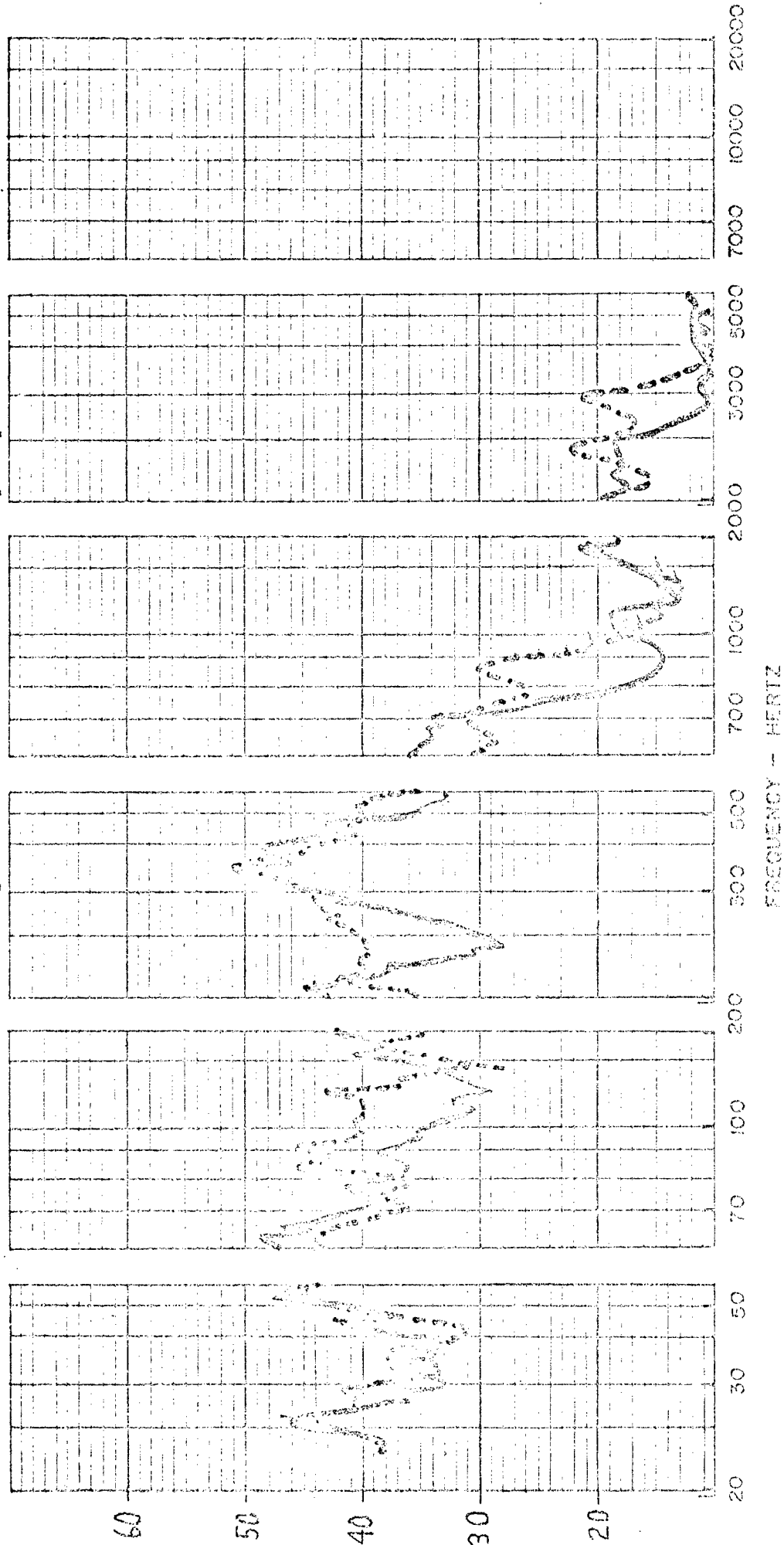


FIGURE 3.1.4

SOUND 3 FEET FROM AIR CLEANERS WITH ANECHOIC  
TERMINATION ON INLETS (RADIATED SOUND)

— Donaldson unit  
- - - Farr unit

6% bandwidth sound pressure level in dB versus frequency



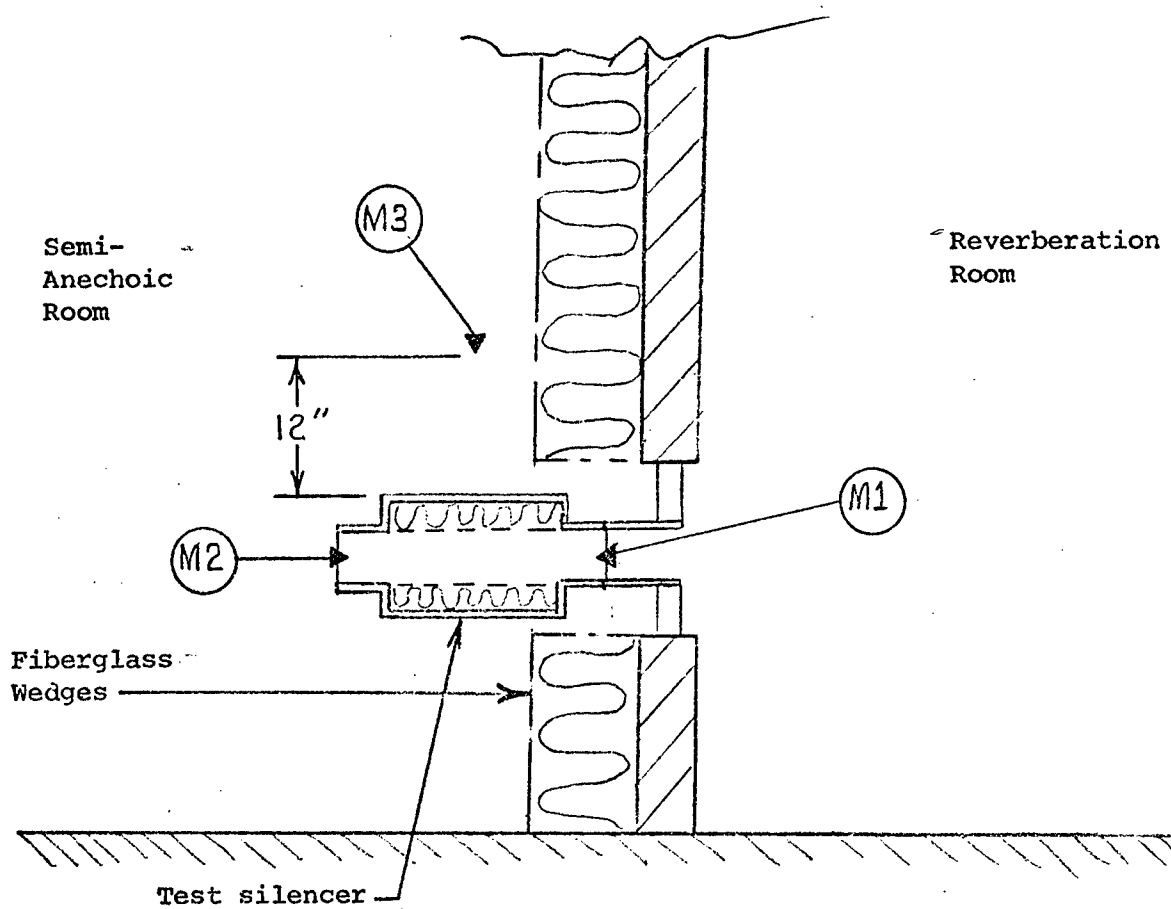


FIGURE 3.1.5

NO FLOW AIR INTAKE SILENCER TEST FIXTURE

M1, M2 and M3 are microphone positions



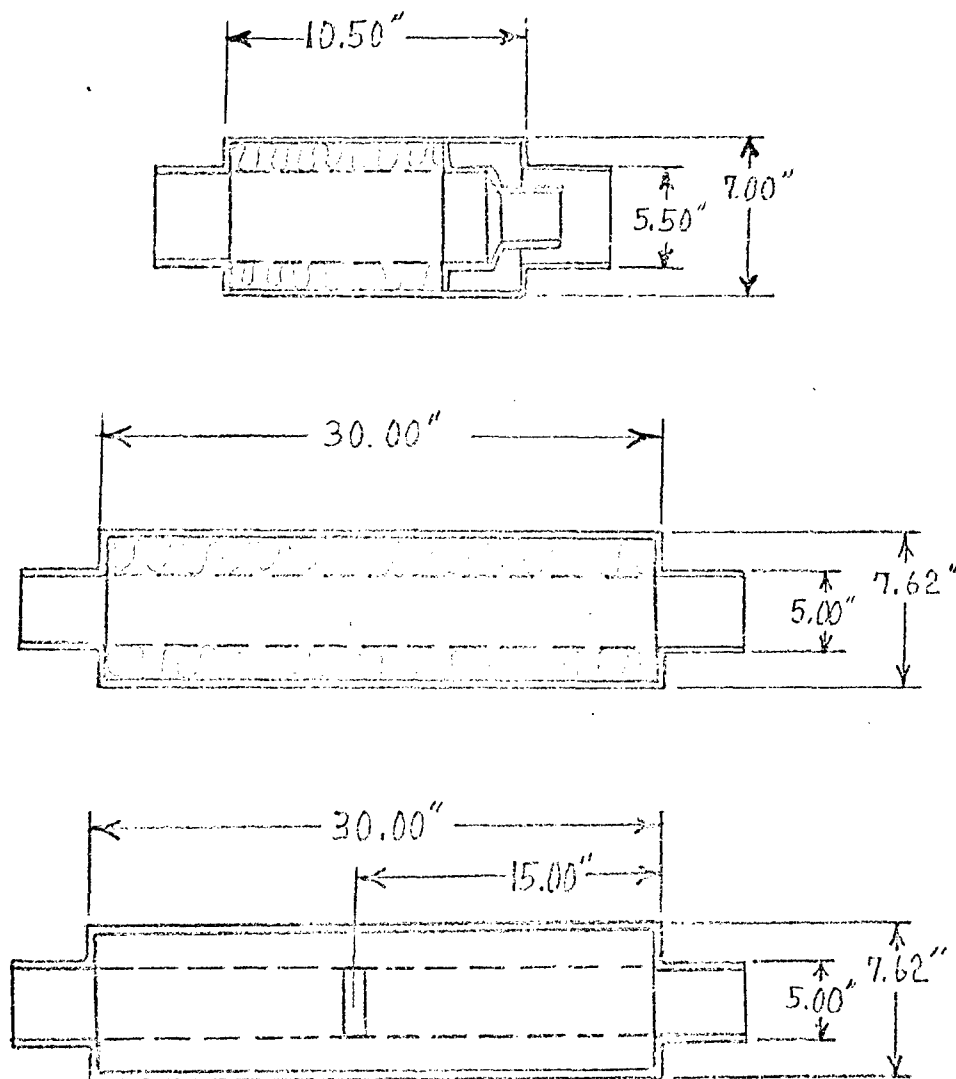


FIGURE 3.1.6

# SECTION VIEWS OF THREE SILENCERS

Top - Donaldson Silencer  
Center - Lined Nelson Silencer  
Bottom - Baffle Nelson Silencer

Figure 3.1.7

BAFFLE SILENCER UNCORRECTED ATTENUATION  $\square$  —  $\square$   
 36" PIPE ATTENUATION  $\circ$  —  $\circ$

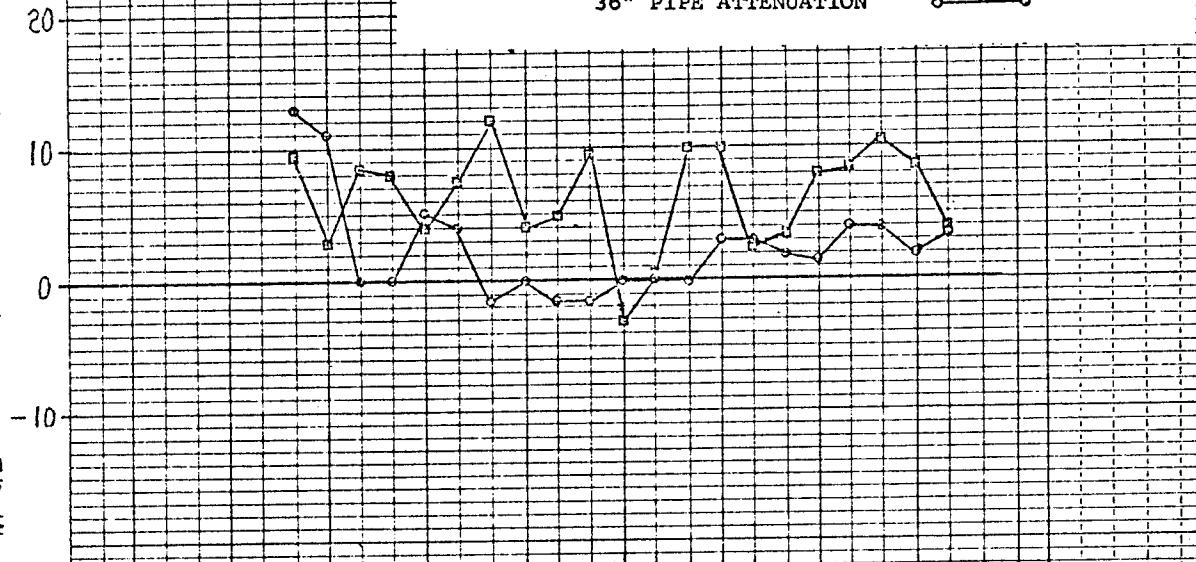


Figure 3.1.8

BAFFLE SILENCER CORRECTED ATTENUATION

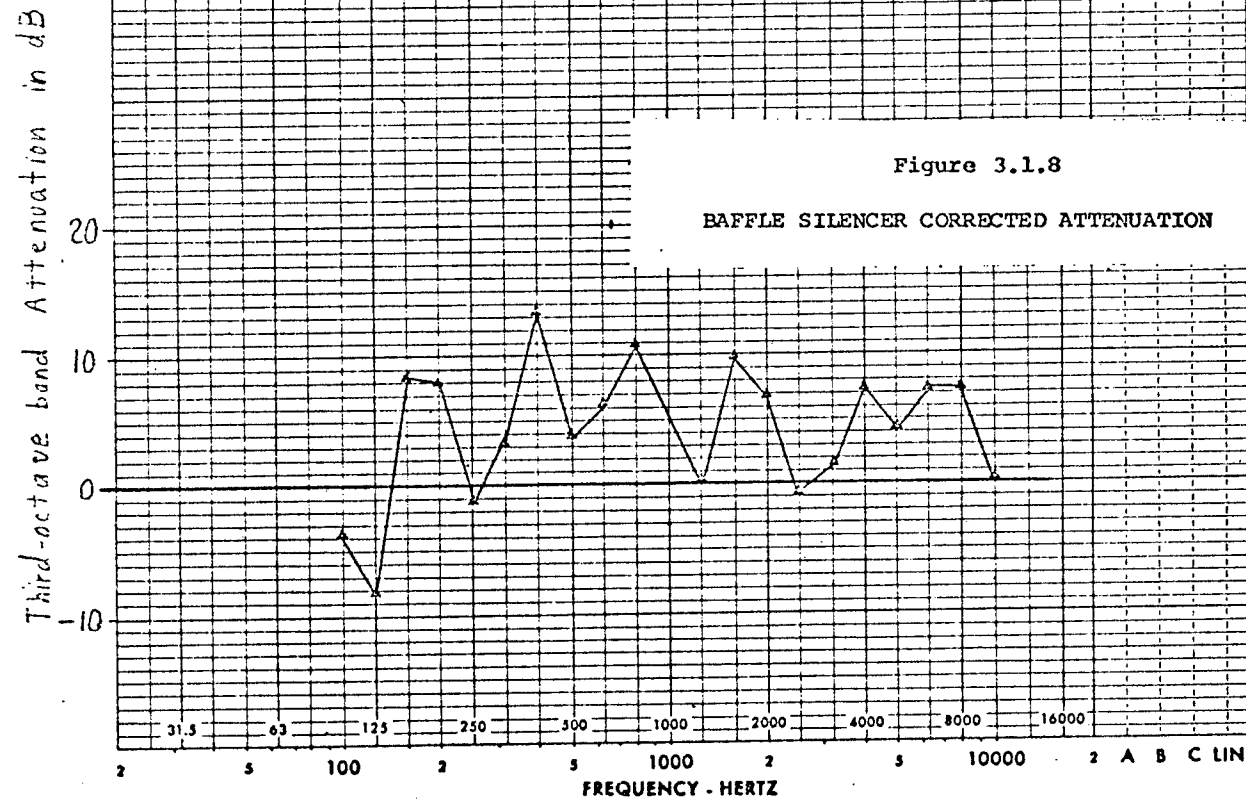
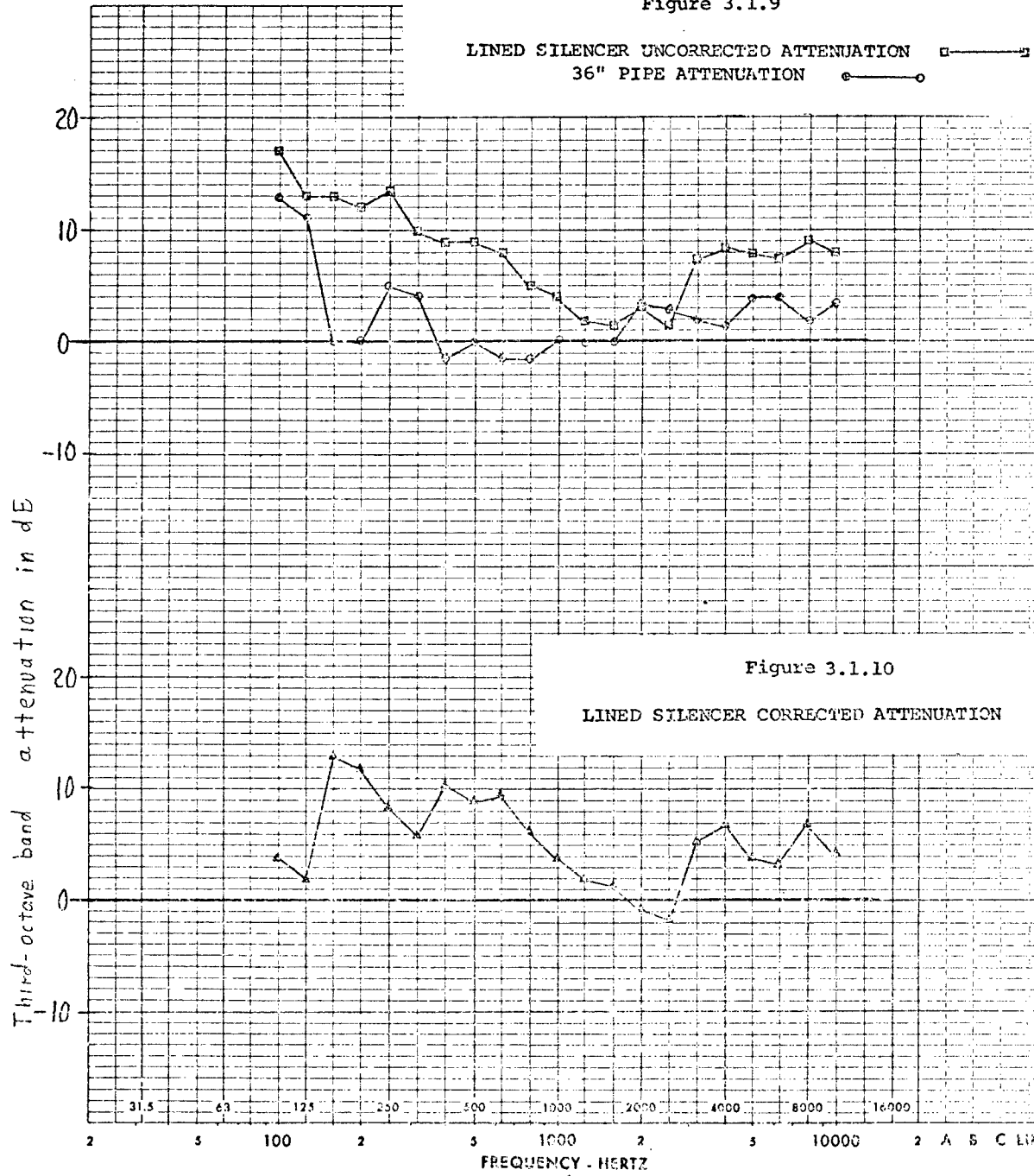


Figure 3.1.9



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Figure 3.1.11

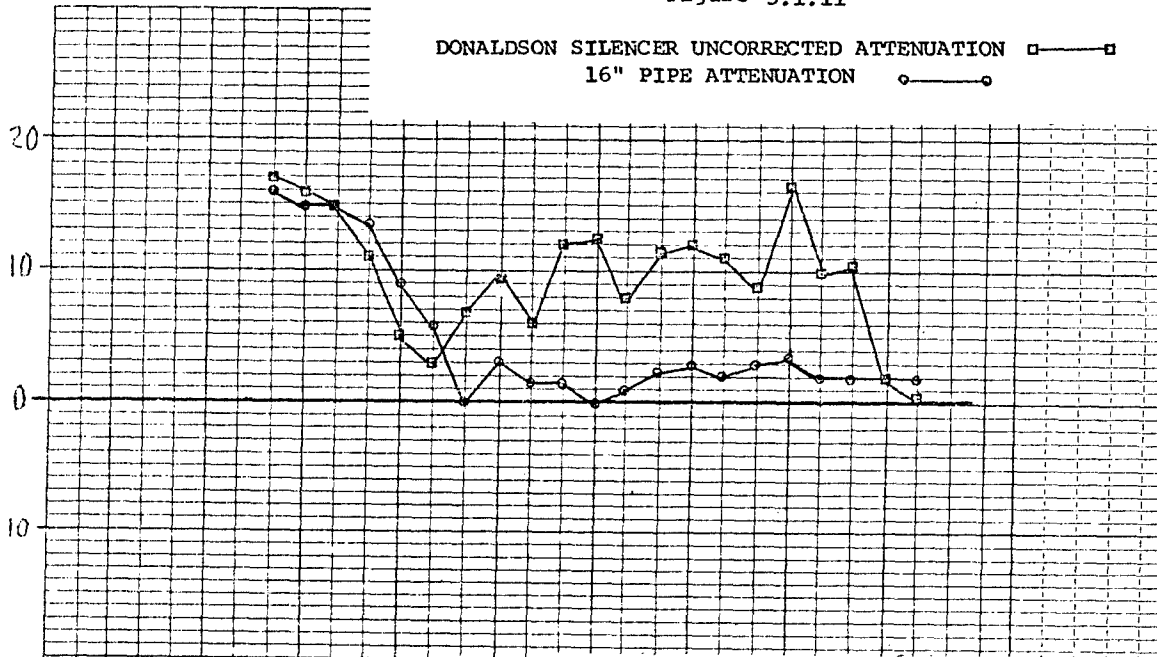


Figure 3.1.12

DONALDSON SILENCER CORRECTED ATTENUATION

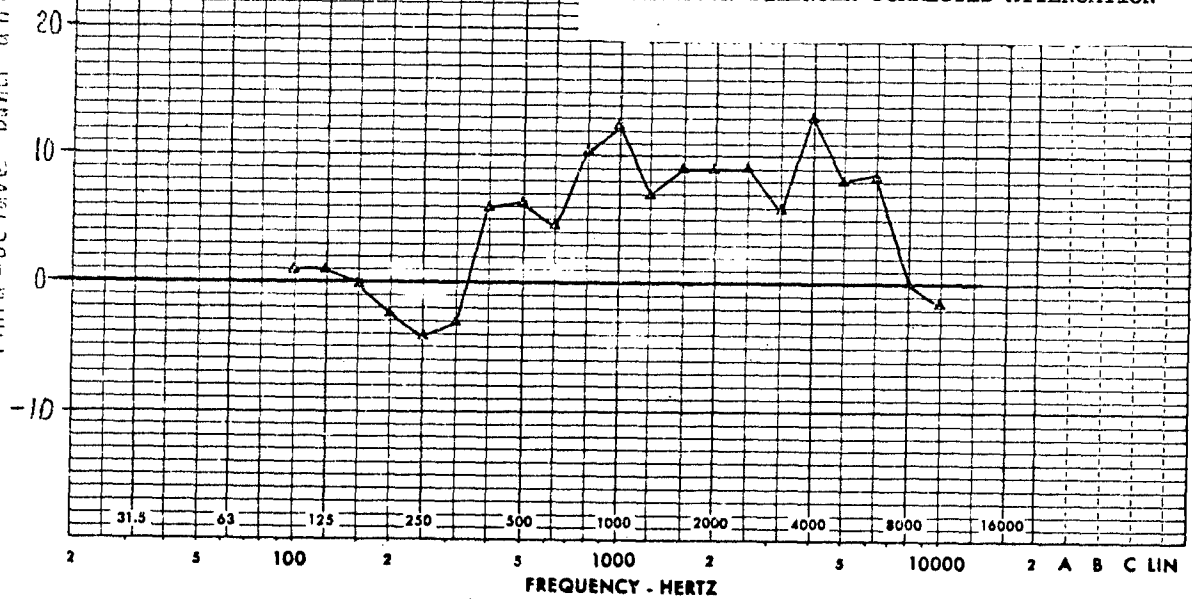
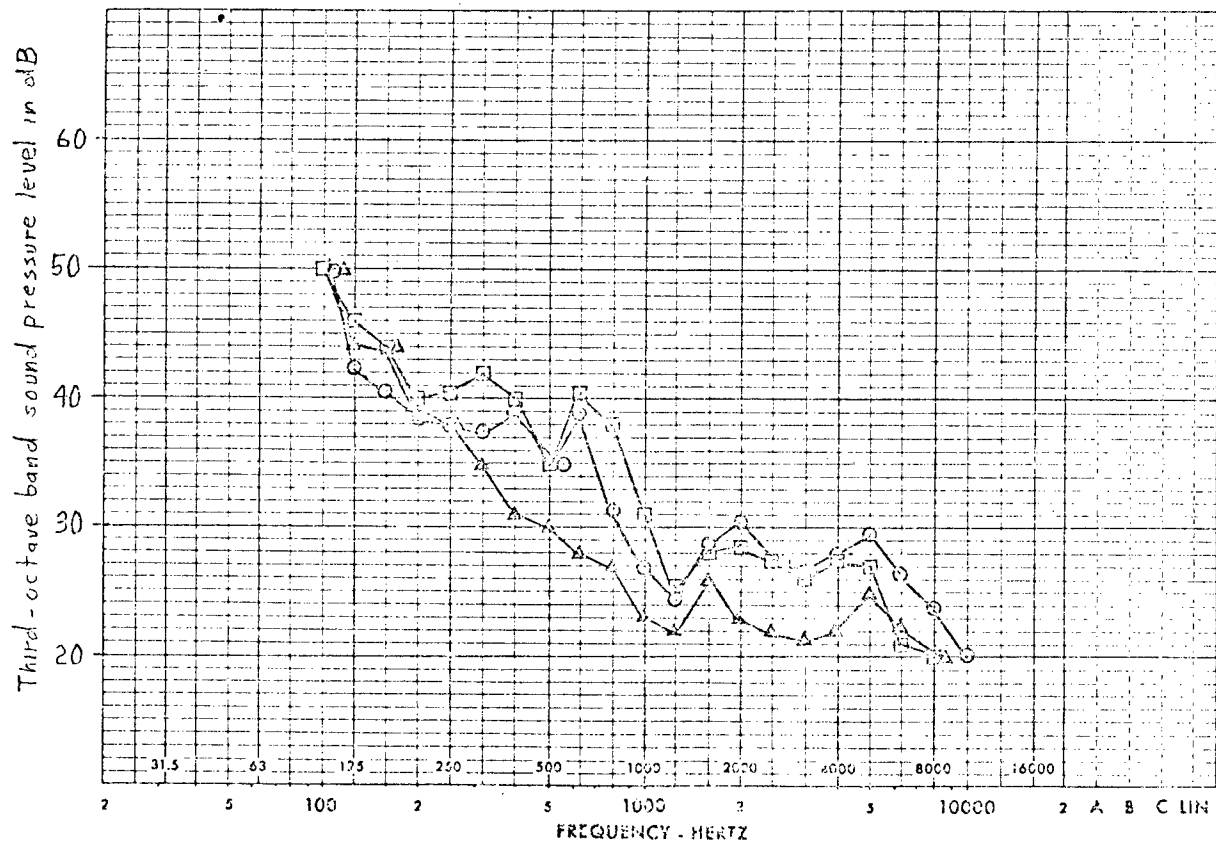


FIGURE 3.1.13

RADIATED NOISE FROM THREE INTAKE SILENCERS

No flow test with anechoic termination sealing intake end of silencer. Microphone 12" to side of silencer.

- — □ Lined Nelson Silencer
- — ○ Baffle Nelson Silencer
- △ — △ Donaldson Silencer



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FIGURE 3.2.1

FAN NOISE AT THE OPERATOR'S EAR POSITION  
FANS DRIVEN BY QUIET ELECTRIC MOTOR AT 1080 RPM

- Background, motor running
- ◇—◇ Stock fan and shroud
- △—△ Large blade fan
- Ring fan

Octave band or weighted sound level in dB versus frequency

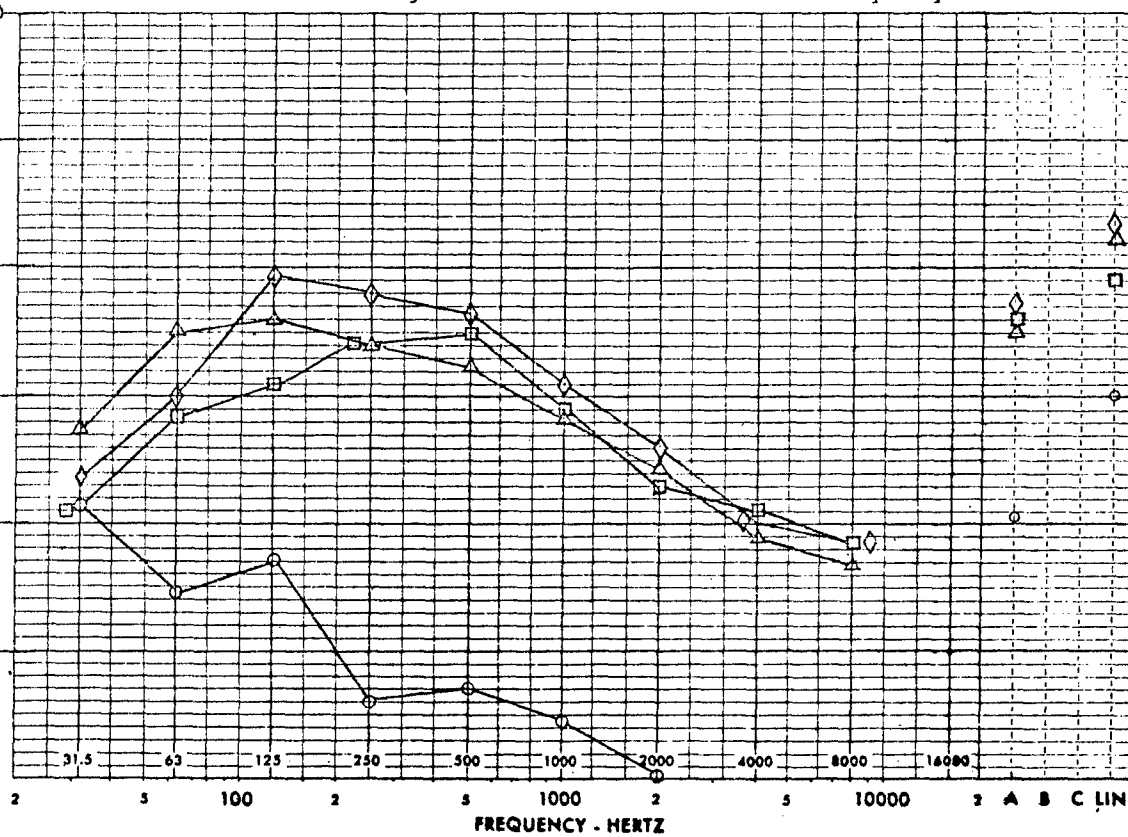


FIGURE 3.2.2

FAN NOISE 10' FROM ENGINE FAN  
FANS DRIVEN BY QUIET ELECTRIC MOTOR AT 1080 RPM

- Background, motor running
- ◇—◇ Stock fan and shroud
- △—△ Large blade fan
- Ring fan

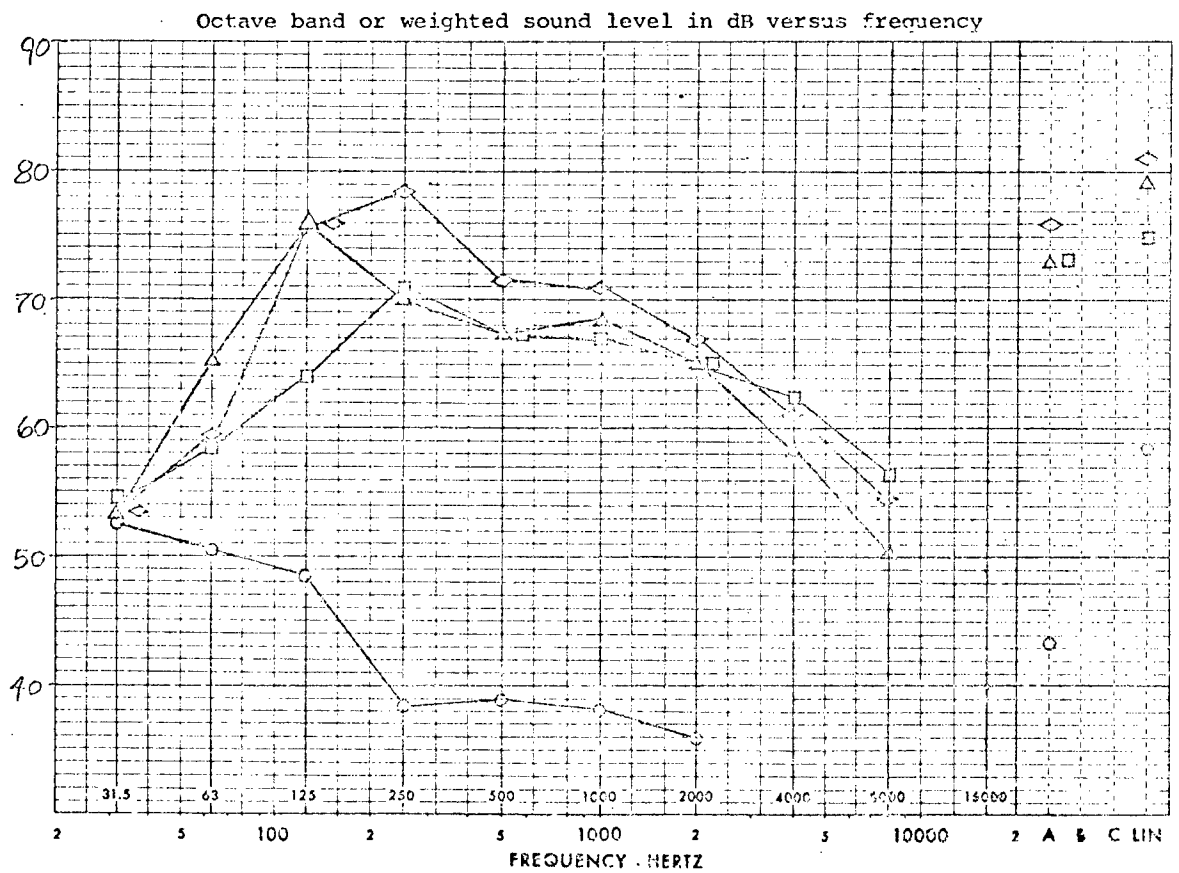


FIGURE 3.2.3

FAN NOISE IN ENGINE COMPARTMENT  $2\frac{1}{2}$ " FROM POWER STEERING PUMP  
FANS DRIVEN BY QUIET ELECTRIC MOTOR AT 1080 RPM

- ◇ Stock fan and shroud
- △ Large blade fan
- Ring fan

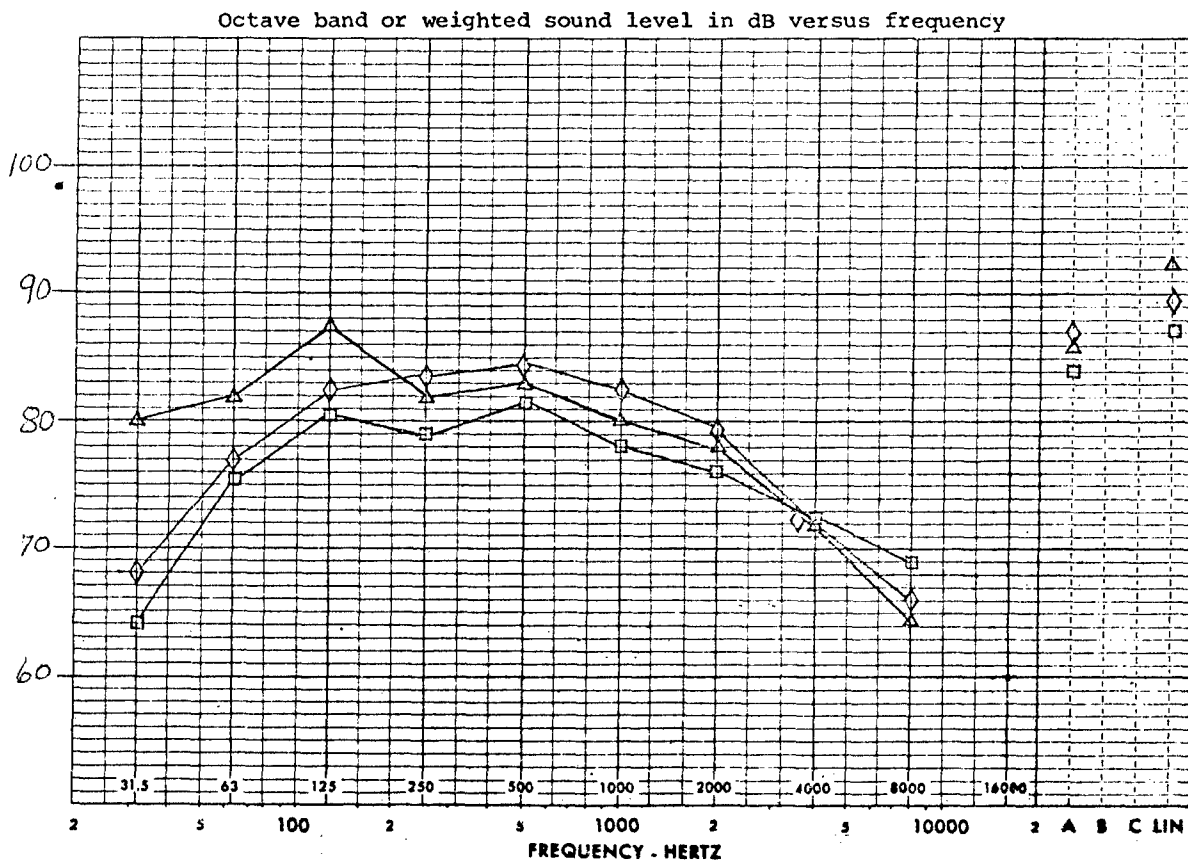




FIGURE 3.2.4

FAN NOISE 10 FT. FROM STOCK "PUSHER" FAN  
DRIVEN AT 1080 RPM BY QUIET ELECTRIC MOTOR

Fan mounted on engine in vehicle

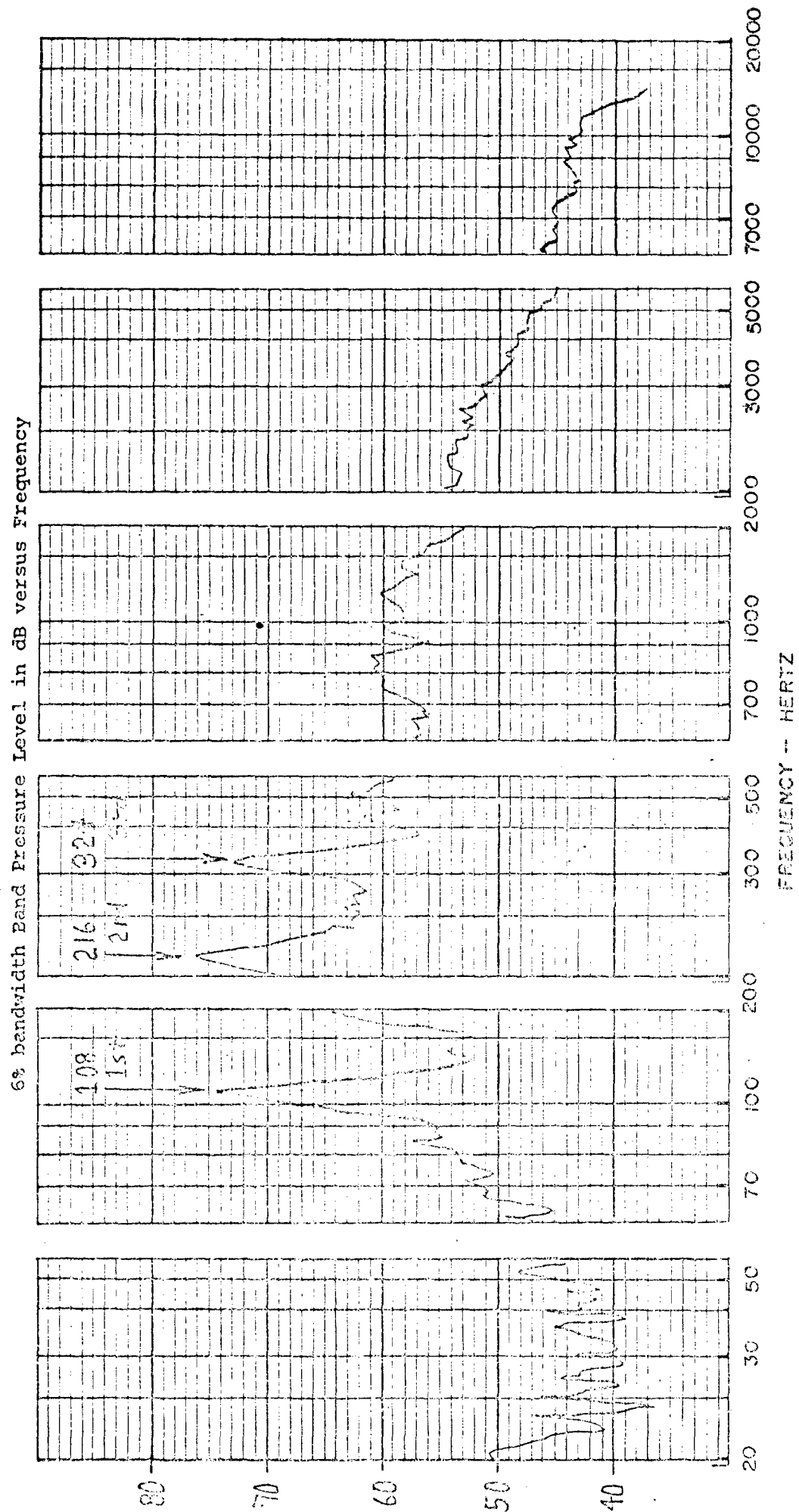


FIGURE 3.2.5

FAN NOISE 10 FT. FROM LARGE BLADE CONVENTIONAL  
FLOW FAN DRIVEN AT 1080 RPM BY QUIET ELECTRIC MOTOR

Fan mounted on engine in vehicle

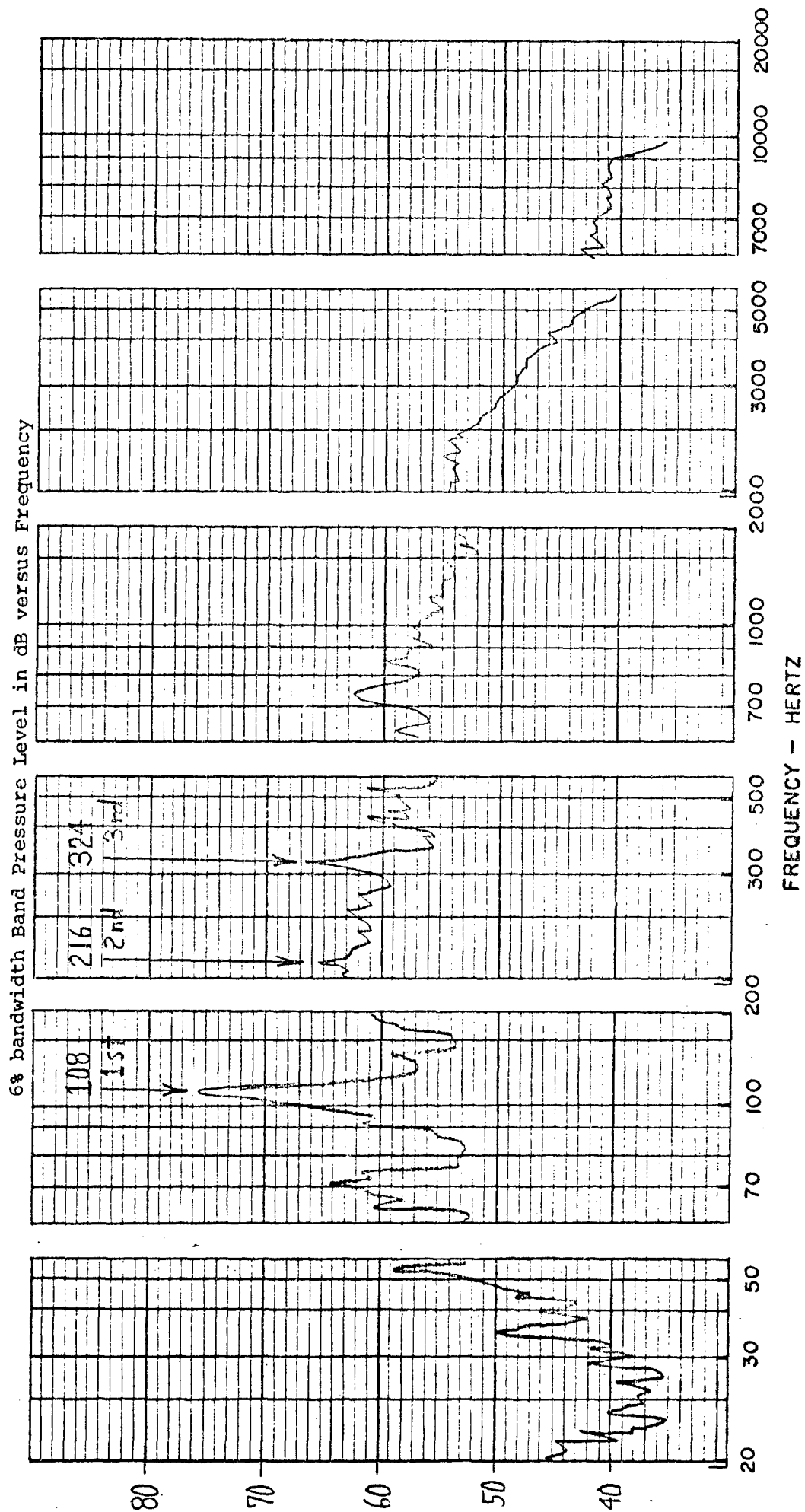


FIGURE 3.2.6

FAN NOISE 10 FT. FROM RING SHROUD "PUSHER" FAN  
DRIVEN AT 1080 RPM BY QUIET ELECTRIC MOTOR

Fan mounted on engine in vehicle

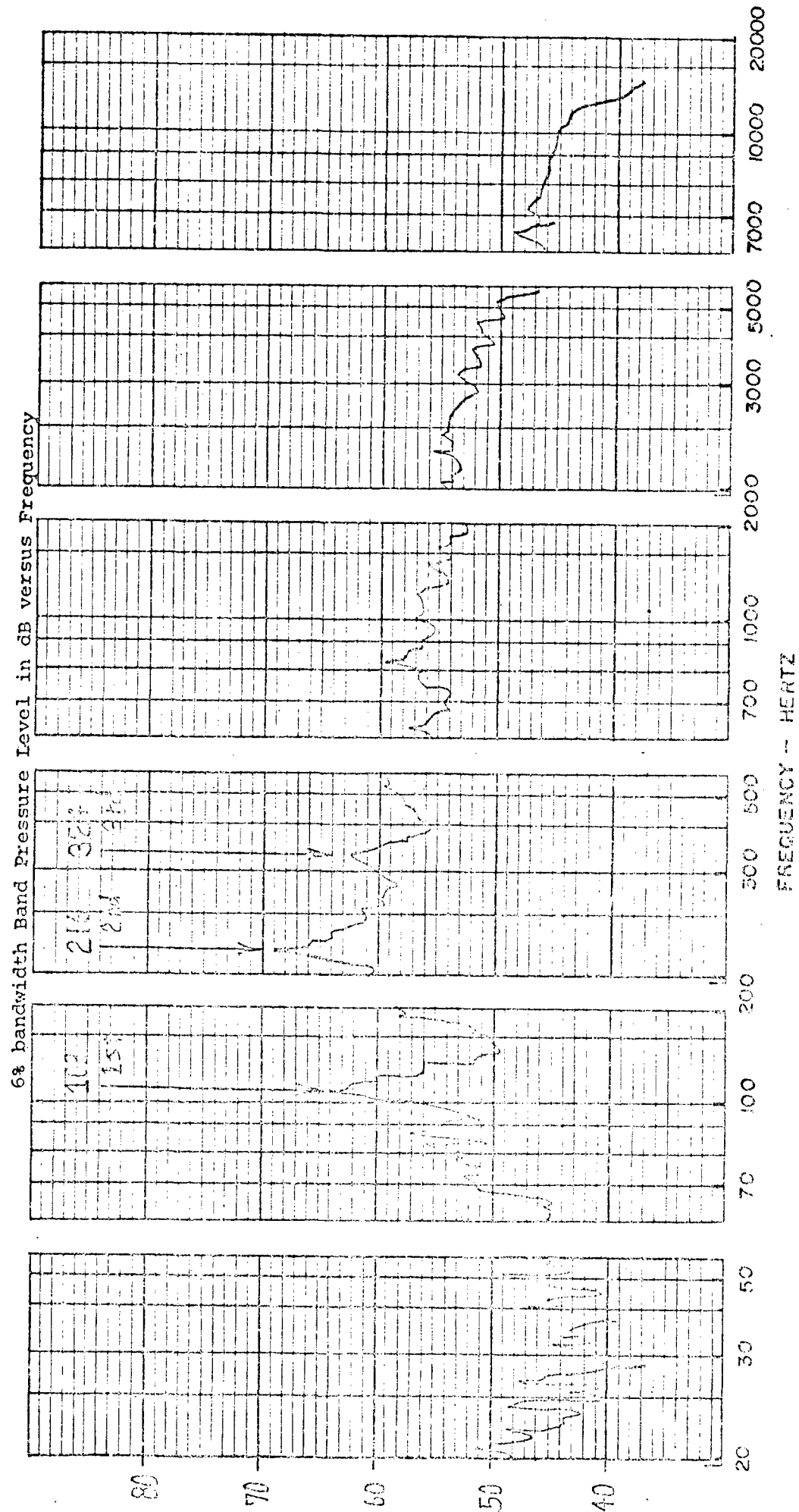


FIGURE 3.3

SOUND 6" BELOW THE MAIN HYDRAULIC LIFT PUMP AND HYDRAULIC BRAKE PUMP

- ▼—▼ No lift or brake pump (from Figure 2.20). High idle condition.
- Warner-Motive lift and brake pumps (from Figure 2.20). High idle condition.
- Commercial Shearing lift pump and Warner-Motive brake pump. High idle condition.
- Commercial Shearing lift pump and Warner Motive brake pump. High idle lifting 8000 lb. load.

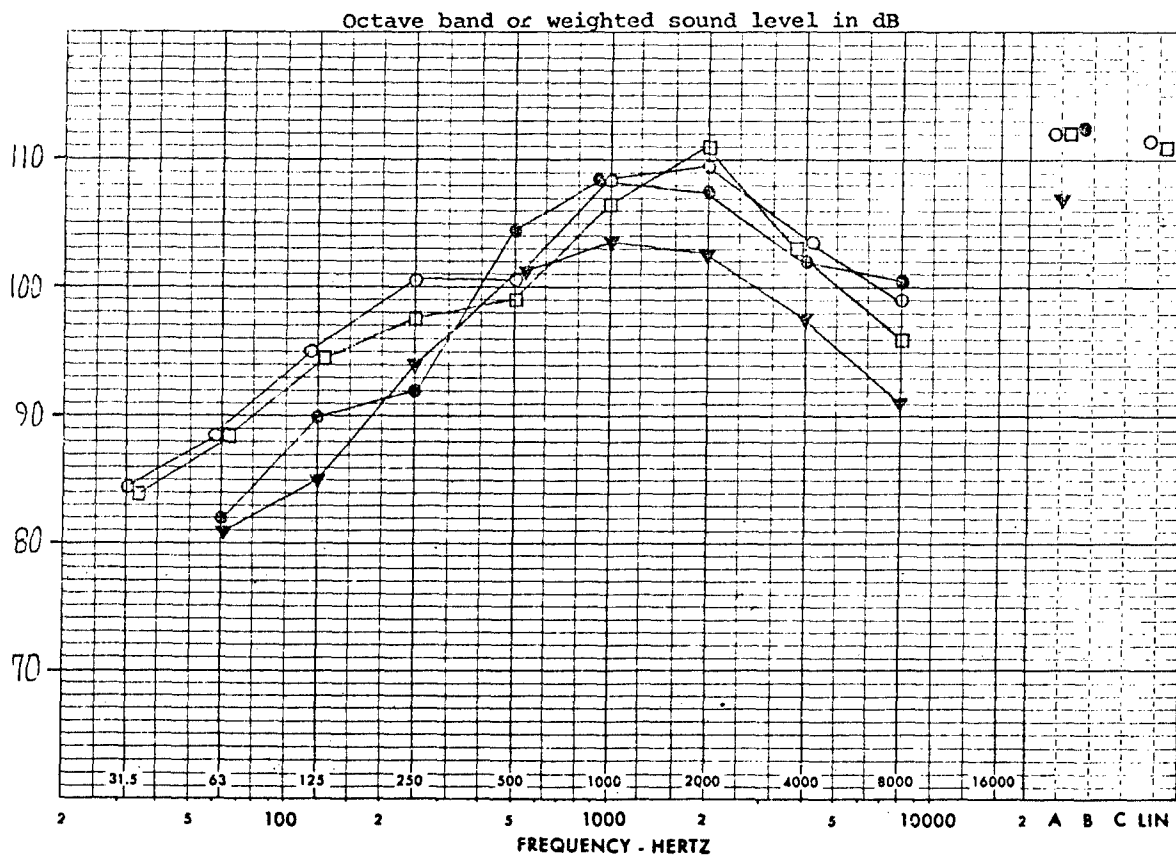


FIGURE 3.4.1

SOUND 6" FROM LEFT EXHAUST OUTLET  
HIGH IDLE CONDITION

Note: All exhaust systems were dual (twin muffler) systems.

- Original muffler
- Wrapped Stemco muffler
- ▽—▽ Wrapped Riker muffler

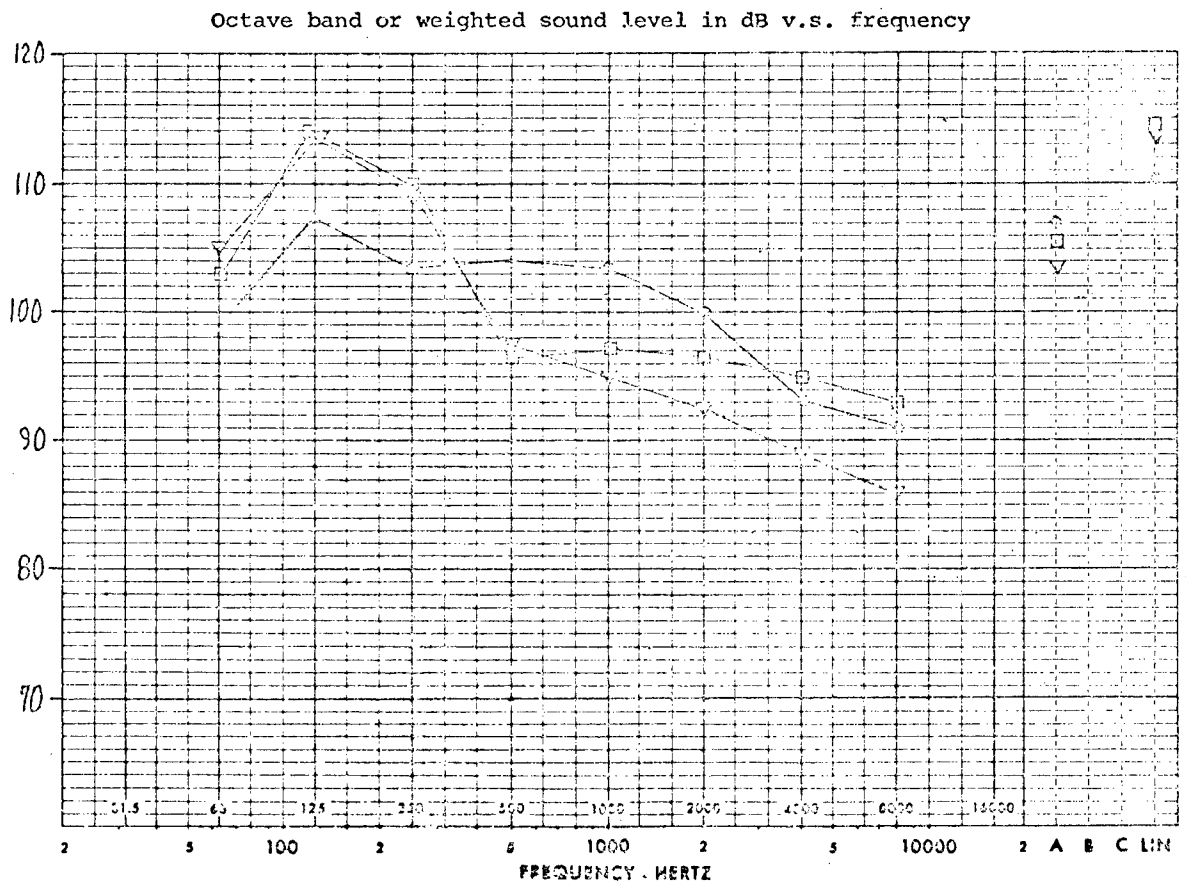


FIGURE 3.4.2

SOUND 6" FROM LEFT EXHAUST OUTLET  
HIGH IDLE CONDITION

Note: All exhaust systems were dual (twin muffler) systems.

- Original muffler
- ◇—◇ First wrapped Nelson muffler
- △—△ Second wrapped Nelson muffler

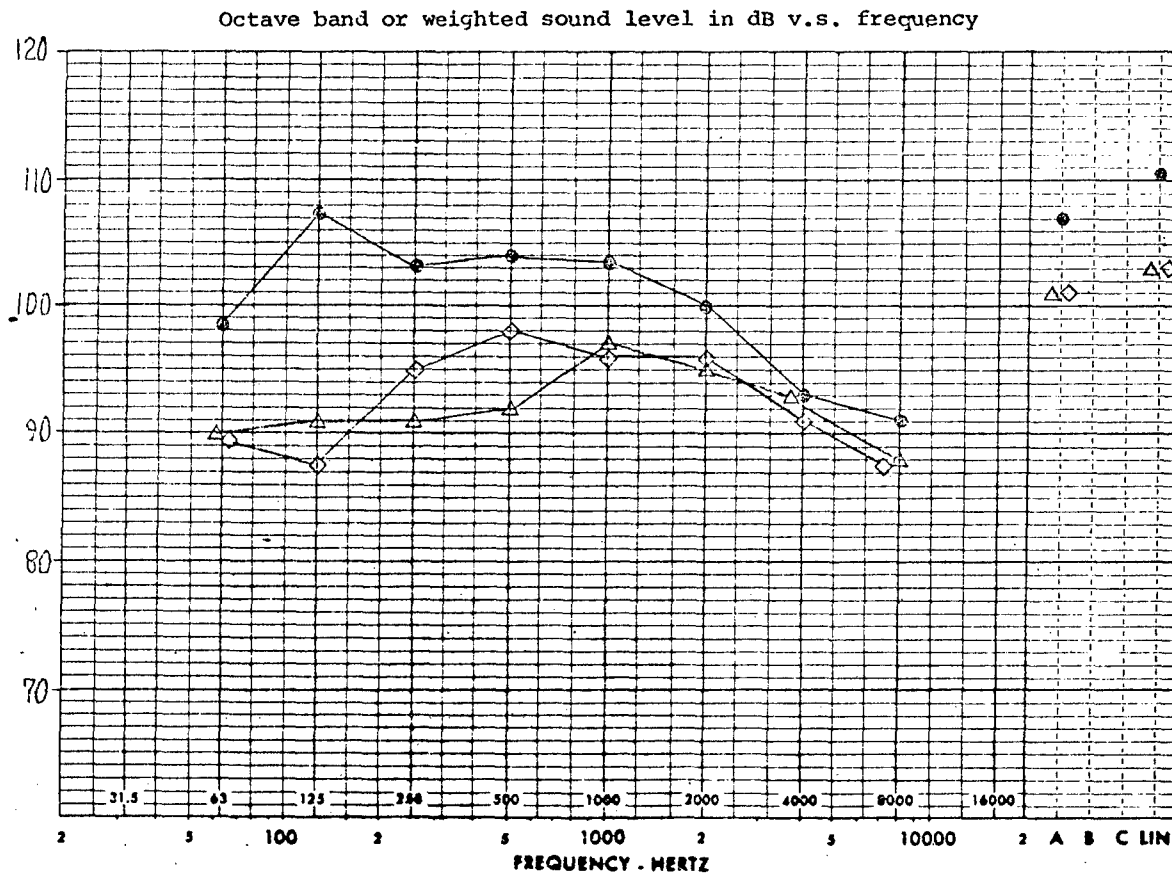


FIGURE 4.1.1

90 DB(A) PACKAGE COMPONENTS SUMMARIZED

- a) Enclosure Parts: Fourteen sheet metal parts to close existing holes or holes created when components were relocated. One flexible part of limp barrier material on upper boom.
- b) Damping Treatment: Used on the hood, lift control lever panel, and panel by the operator's right leg.
- c) Absorption Treatment: Eight vinyl clad pads of 1" thick urethane foam in the engine compartment.
- d) Intake: Relocated air cleaner to right rear fender. Damping on cleaner. Added silencing element.
- e) Engine Fan: Replaced with lower speed, larger bladed fan. Replaced shroud with closer fitting design.
- f) Exhaust Mufflers: Replaced with special large volume mufflers.
- g) Power Steering Pump: Replaced with quieter pump.
- h) Hydraulic Suction Lines: Rigid lines replaced with hose.
- i) Isolation: Isolated with neoprene mounting pads and snubbing washers with vinyl tubing around the mounting bolts:
  - (1) Engine
  - (2) Transmission
  - (3) Hydraulic Reservoir
  - (4) Hydraulic Control Valves near Operator
- j) Floor Mat: Mat of 1.6 lb. per square foot barrier material bonded to 0.25 inch thick urethane foam.
- k) Seal for Lift Control Levers: a lined duct of vinyl faced urethane foam to allow lever motion while creating a noise seal.

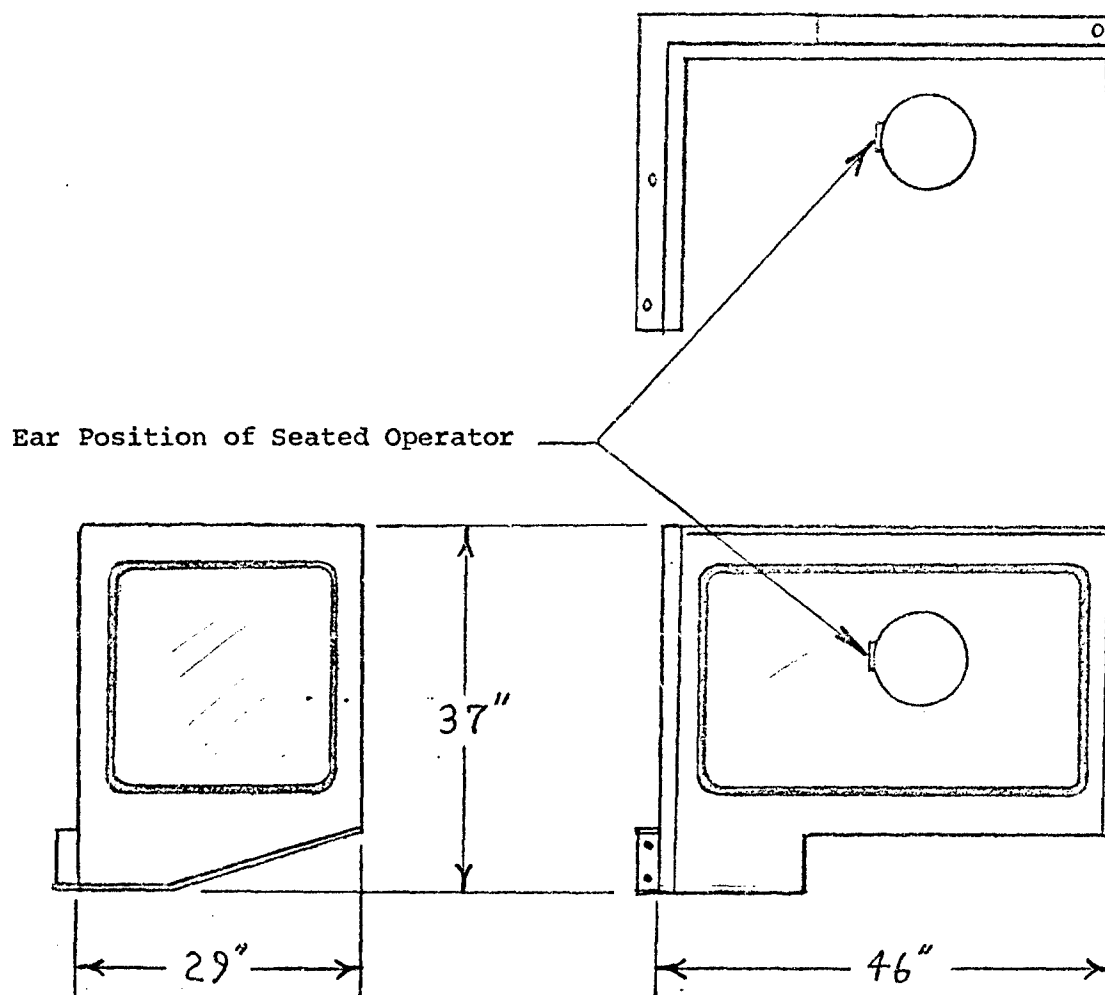


FIGURE 4.1.2

#### SKETCH OF OPERATOR SHIELD

Construction is 12 gage steel with 1/8" safety glass windows. Flexible seals are installed on bottom surfaces which mate to vehicle. Shield extends as far to right of driver as it does to include all lift control levers.



FIGURE 4.2

SOUND AT THE OPERATOR'S EAR POSITION

High Idle Condition

- As received
- 90 dB(A) package
- ◇—◇ 85 dB(A) package

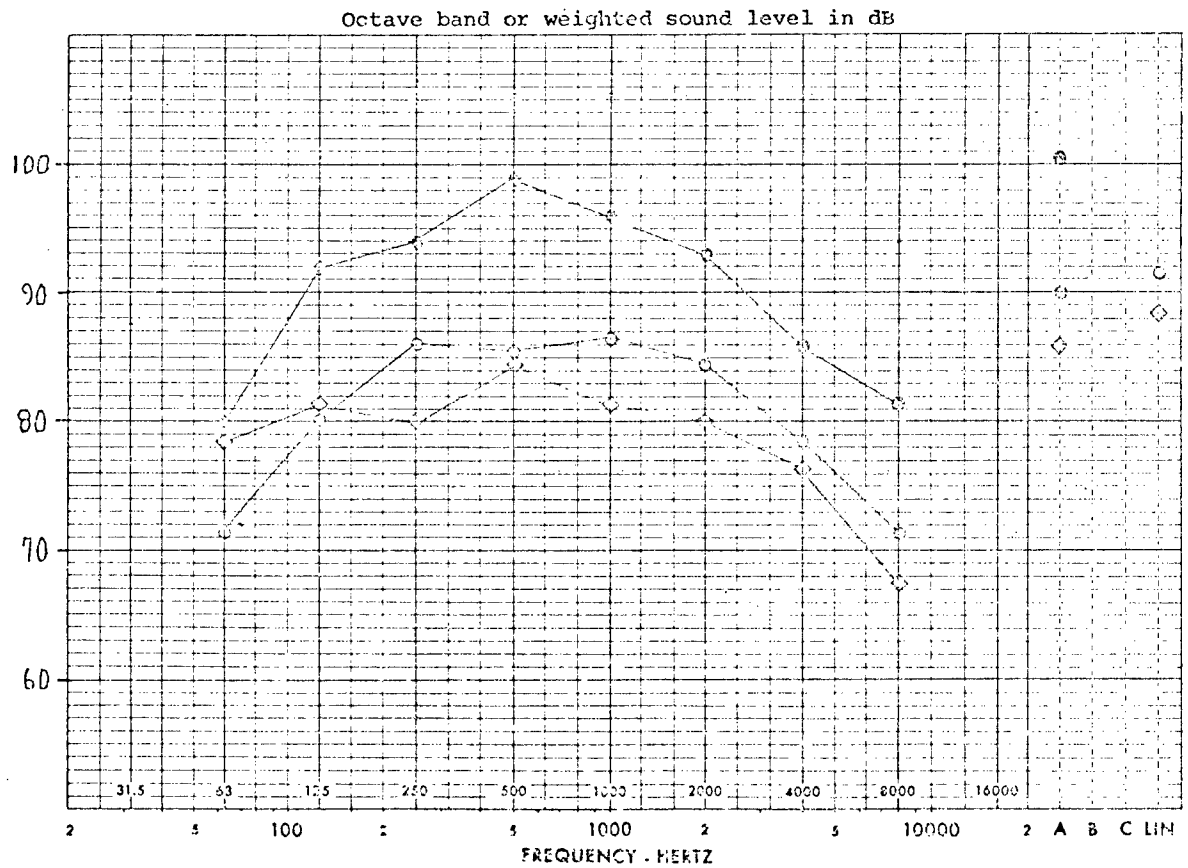


FIGURE 4.3

SOUND AT THE OPERATOR'S EAR POSITION

Torque Converter Stall Condition

- ▲ As received
- 90 dB(A) package
- ◇ 85 dB(A) package

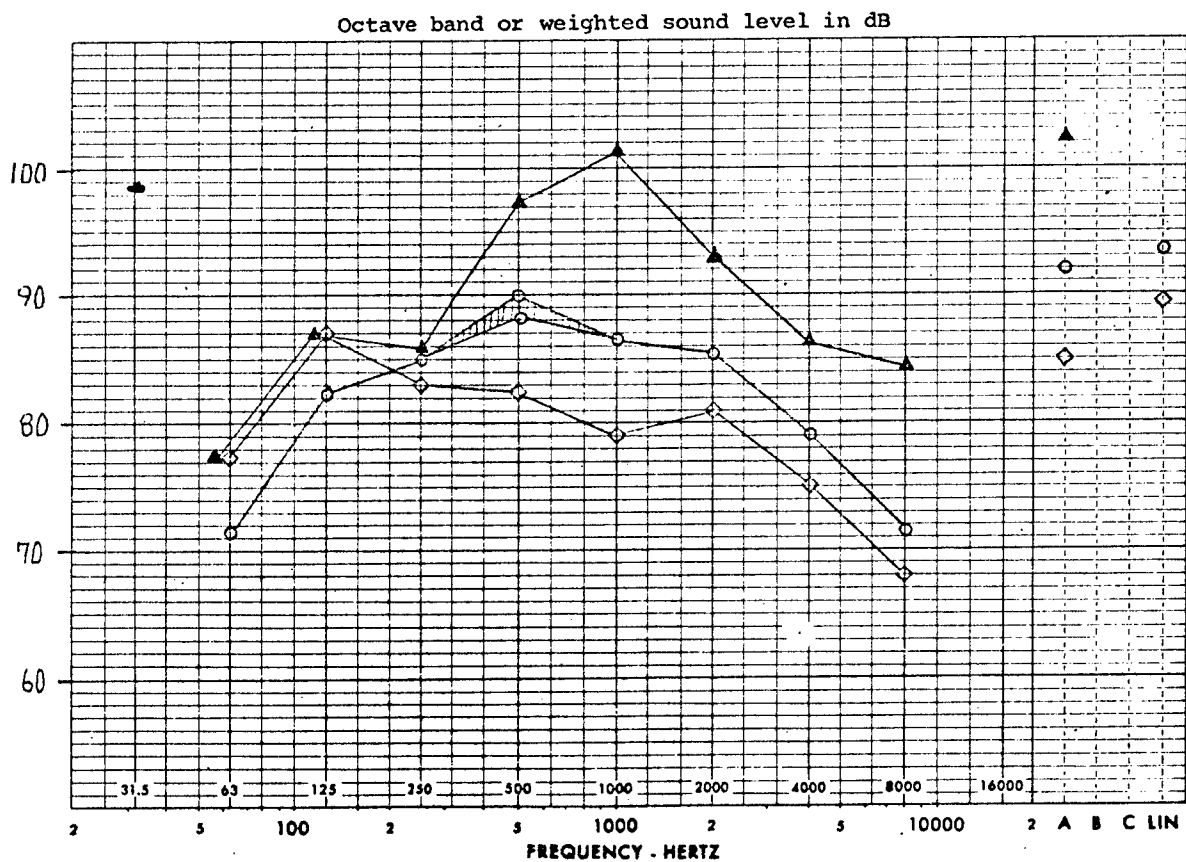


FIGURE 4.4

SOUND AT THE OPERATOR'S EAR POSITION

High Idle Lifting 8000 Lb. Load

- As received
- 90 dB(A) package
- ◇—◇ 85 dB(A) package

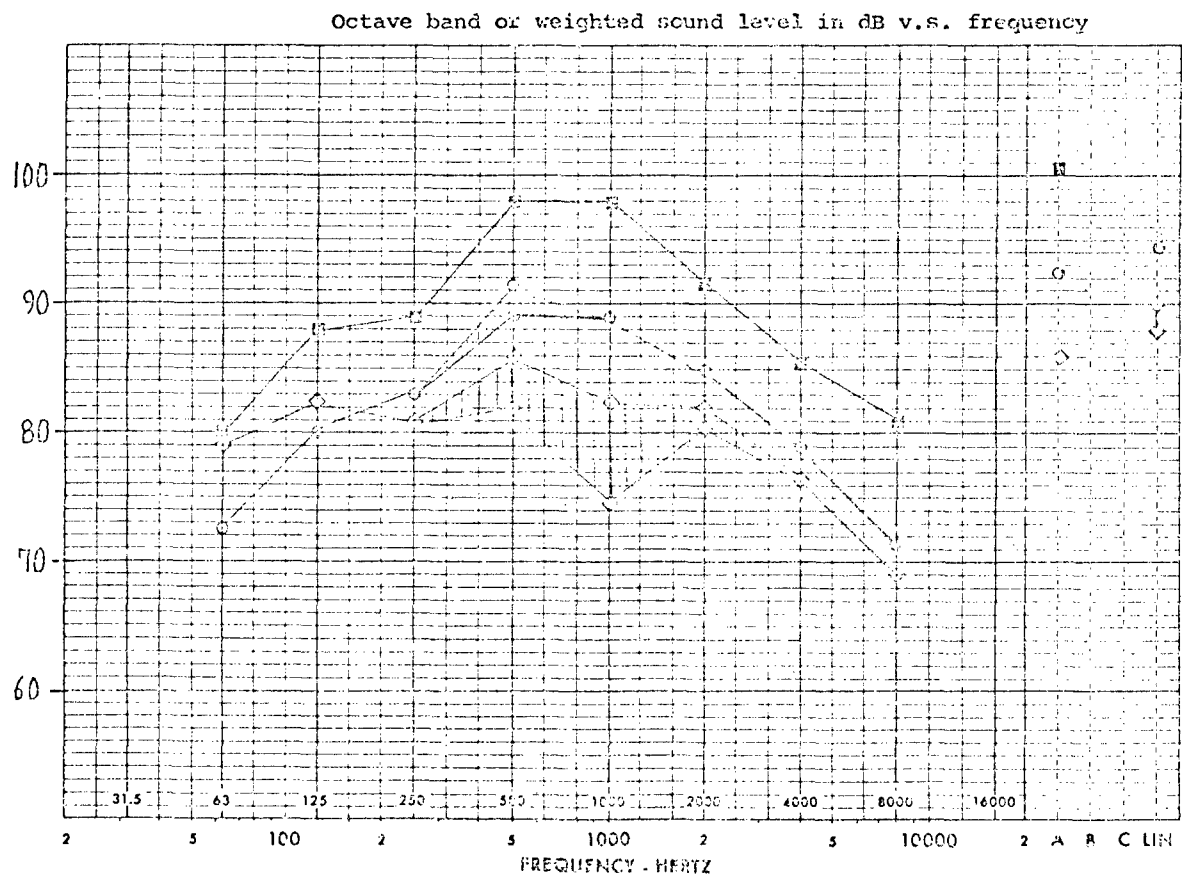


FIGURE 4.5

SOUND IN ENGINE COMPARTMENT

Microphone 8½" from power steering pump

- High idle as received (from Figure 1.13)
- △—△ High idle with treated engine compartment.  
(Tyrone steering pump and Commercial  
Shearing lift pump.)

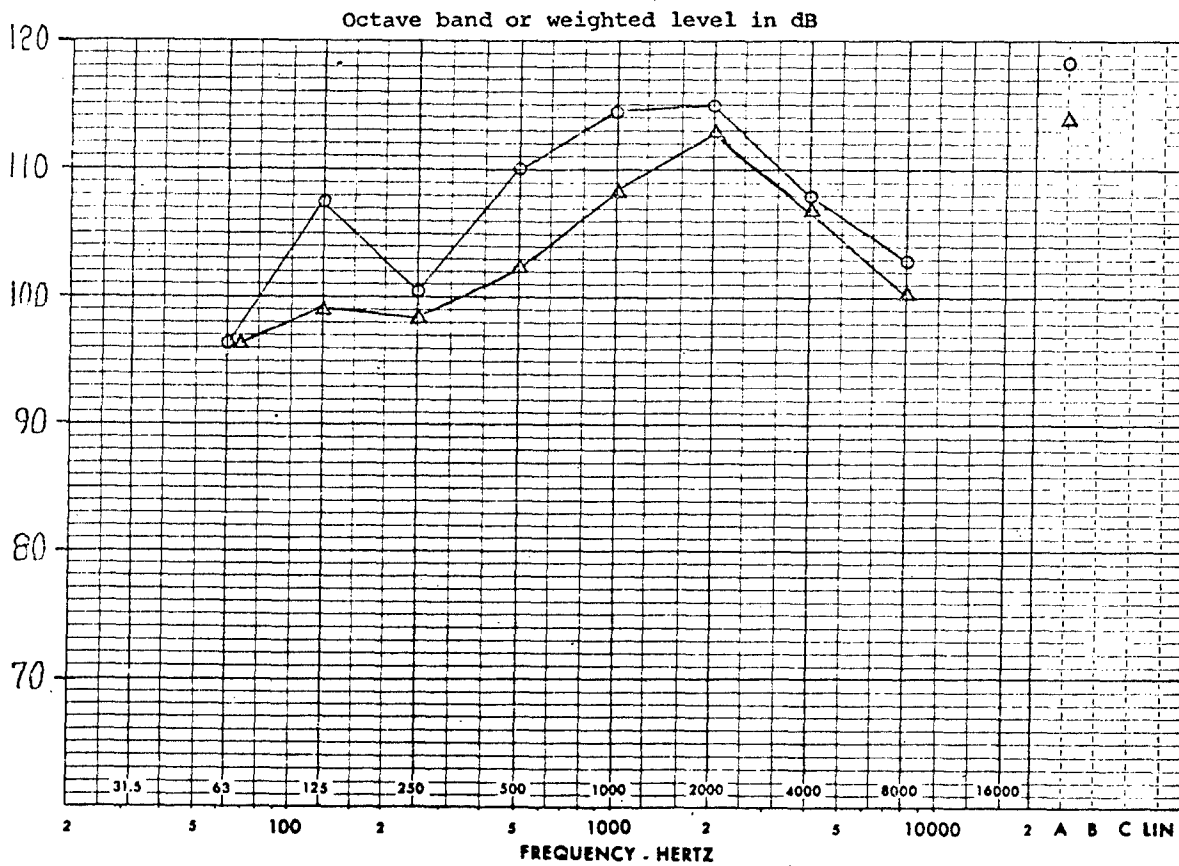
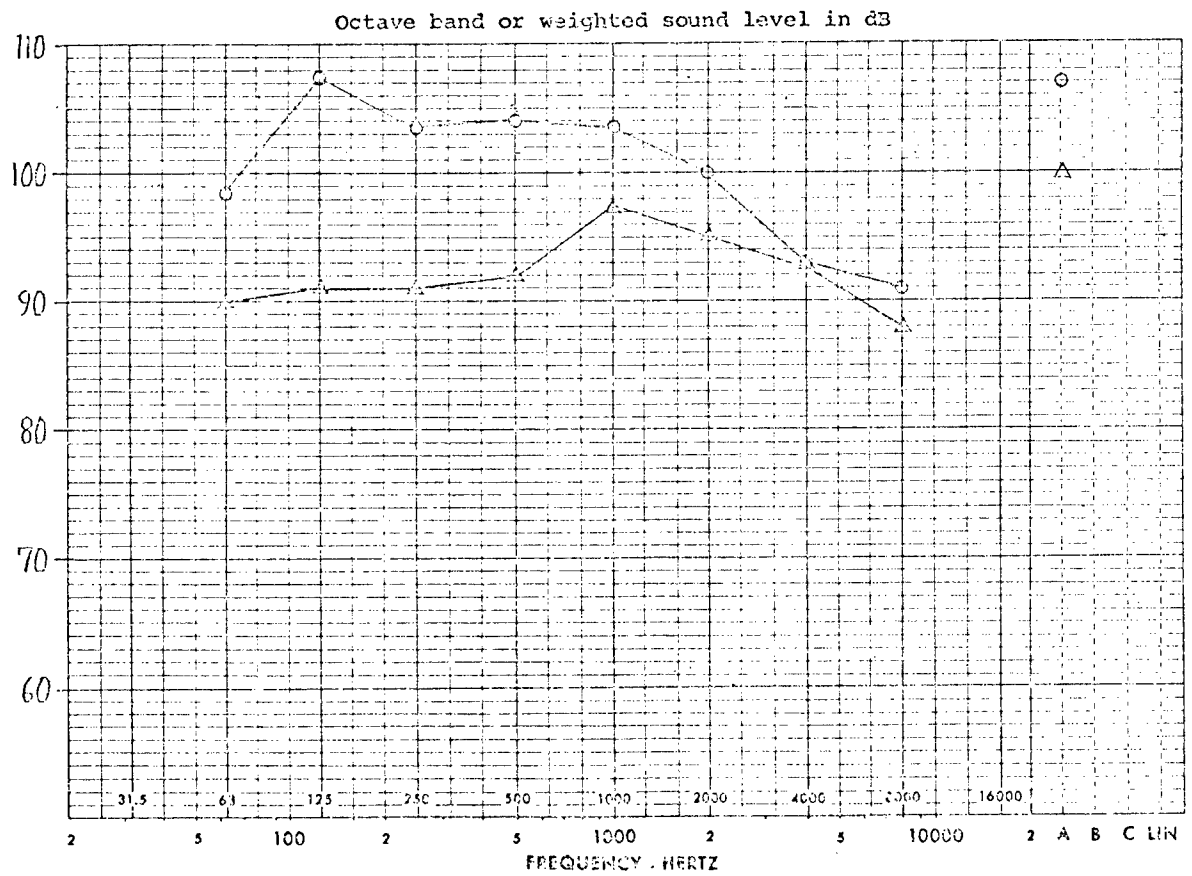


FIGURE 4.6

SOUND 6" FROM LEFT EXHAUST OUTLET

- High idle-as received (from Figure 1.15)
- △—△ High idle-final Nelson mufflers



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FIGURE 4.7

SOUND 6" FROM AIR INTAKE

High idle as received

- High idle. Added Donaldson silencer.  
 △—△ and damped air cleaner, Relocated  
 to right rear fender.

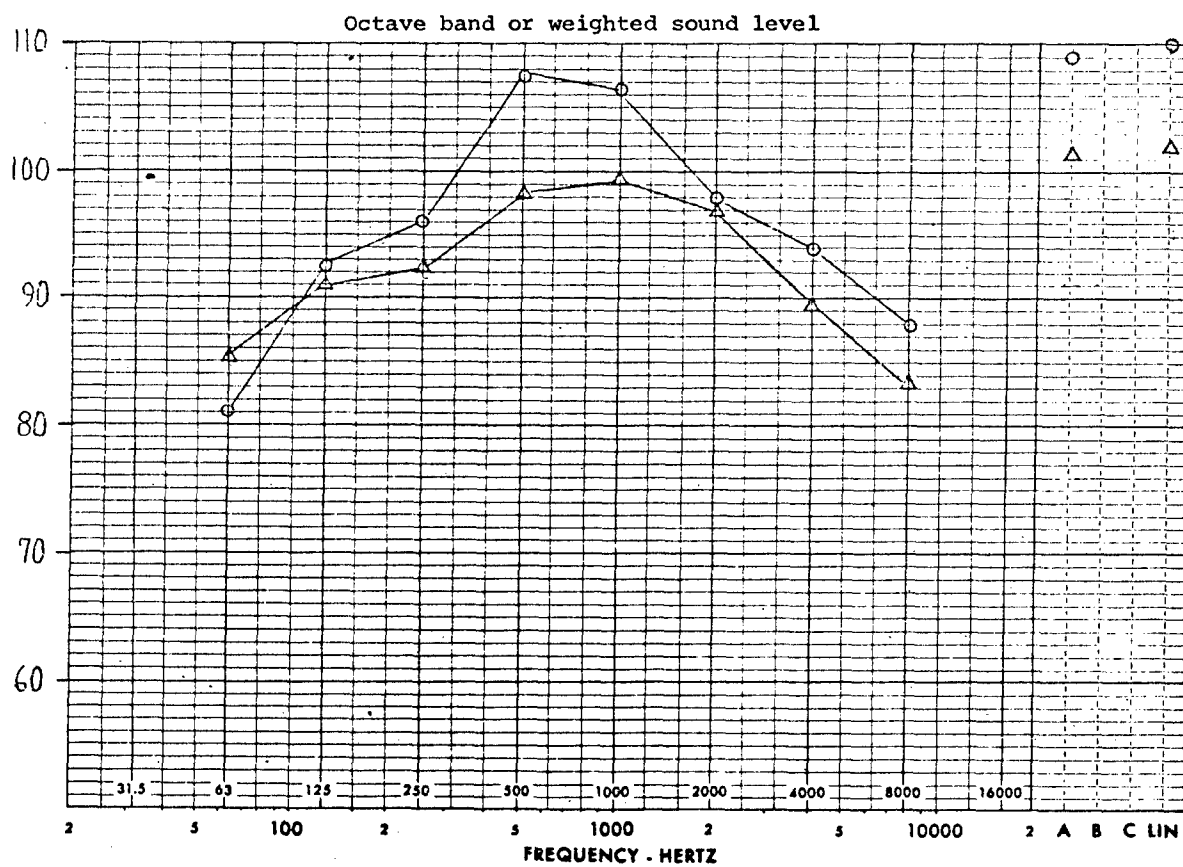


FIGURE 4.8

EFFECT OF EXTRA TREATMENTS ON 90 dB(A) PACKAGE

High Idle Condition-Operator Ear Position

1. —●— 90 dB(A) package
2. —◇— (1) plus sealed top and sides of engine compartment
3. —○— (2) plus top and sides covered with barrier material
4. —△— (3) plus remote exhaust
5. —□— (4) plus fan off

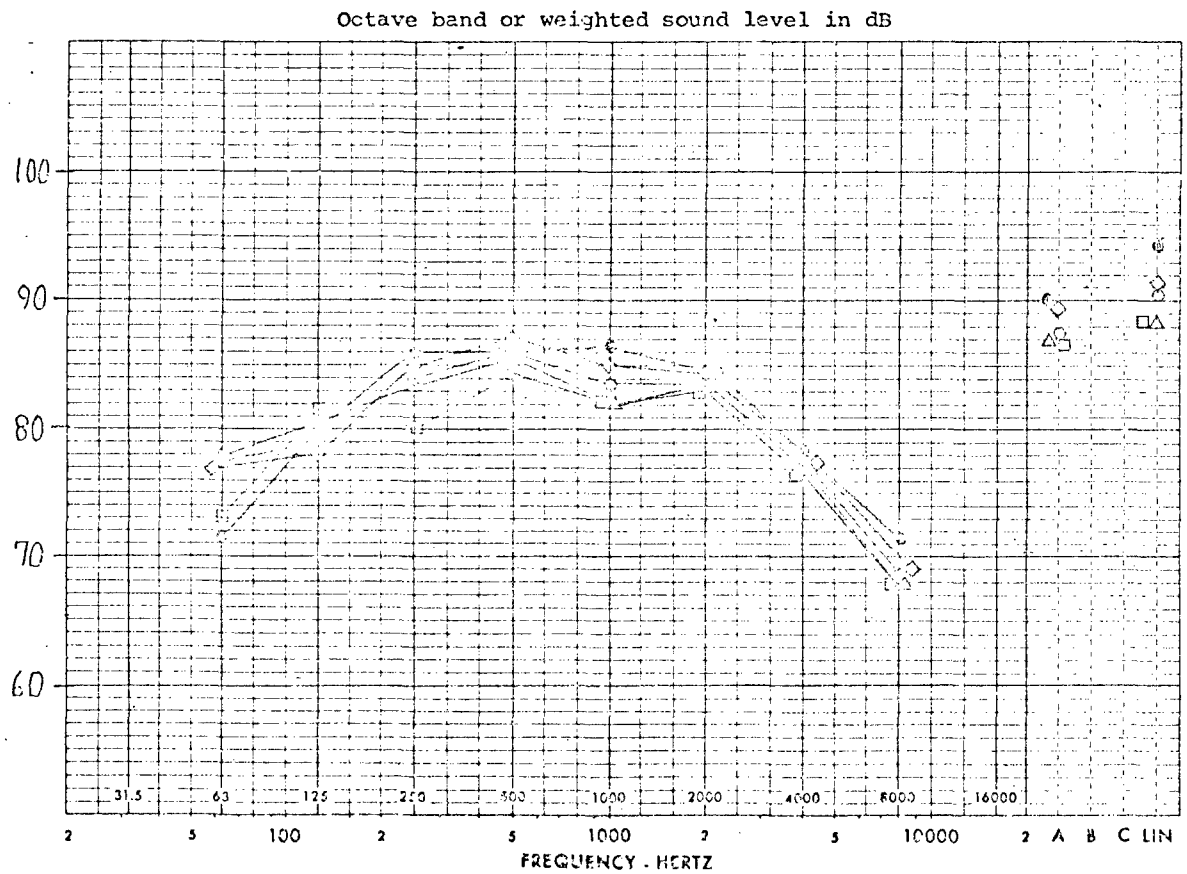


FIGURE 4.9

EFFECT OF REMOVING PROP SHAFT

High Idle Condition-Operator Ear Position

○—○ Prop shaft on  
△—△ Prop shaft off

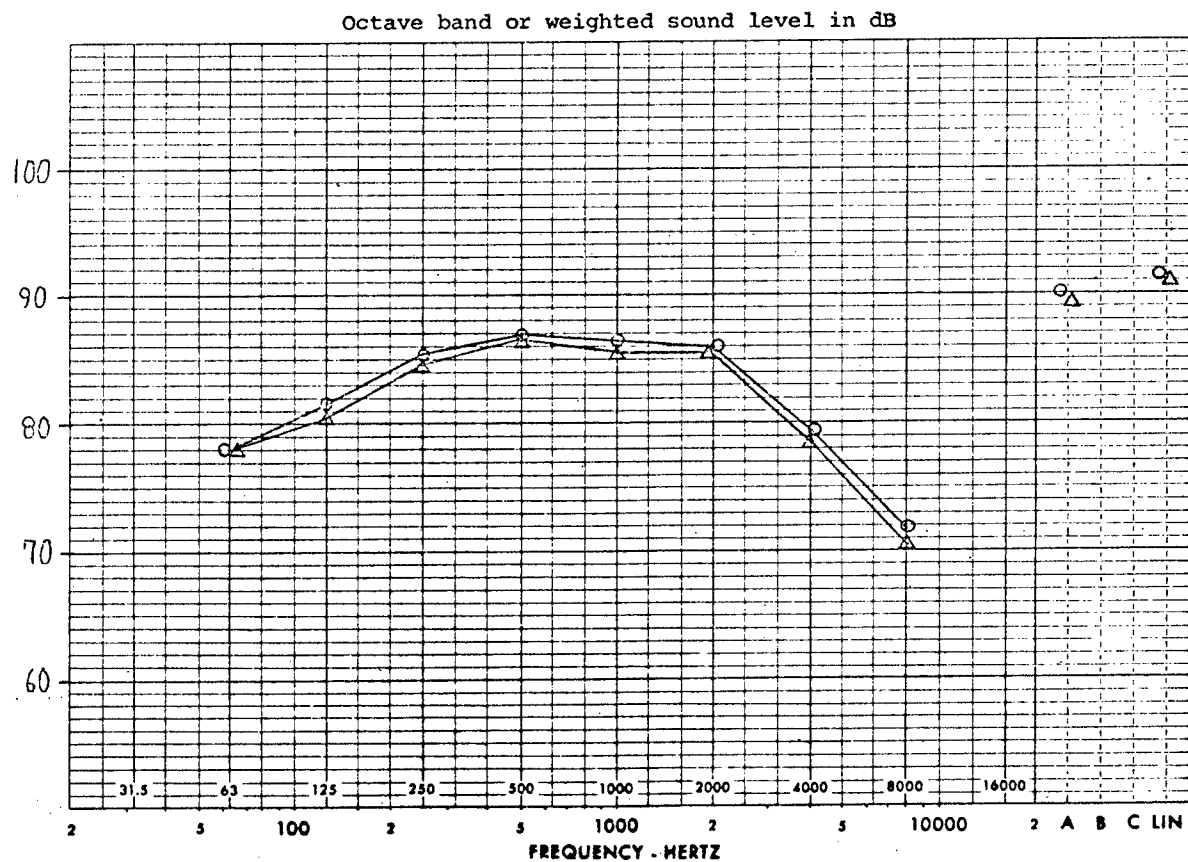




FIGURE 5.1

FINAL PERFORMANCE TEST DATA  
U.S. ARMY ROUGH TERRAIN FORK-LIFT TRUCK  
WITH NOISE REDUCTION KIT

Operation in low gear with four-wheel drive and two-wheel steering. Stable conditions per SAE J872a and SAE J819a. 7000 lb. load on forks.

Date	15 May 1974	
Time	3:00 P.M.	
Track Condition	Dry	
Vehicle weight (LB)	34,500	
Front tire pressure (PSI)	40	
Rear tire pressure (PSI)	45	
Type of fuel used	#2 diesel	
Barometric pressure (mm hg.)	756	
% Relative humidity	74%	
Wind direction	WSW	
Wind velocity	20-30 MPH	
Ambient temperature at track	68°	
Engine speed (RPM)	2700	
Output shaft speed (RPM)	385	
Ground speed (MPH)	3	
Reserve tractive force (LB)	10,500	
Res. tractive HP (calc.)	84	
	T(°F)	ΔT*(°F)
Ambient near vehicle (°F)	68	0
Radiator top tank (°F)	183	115
Radiator bottom tank (°F)	167	99
Engine sump (°F)	218	150
Oil cooler in (°F)	219	151
Oil cooler out (°F)	207	139

\*ΔT is the difference between the given temperature and the ambient near the vehicle.

\*\*All temperatures are averages of stabilized West to East and East to West runs.

FIGURE 5.2

INITIAL AND FINAL FIELD TEST SOUND  
LEVELS AT THE OPERATOR'S EAR

	<u>Original</u>	<u>90 dB(A) Kit</u>	<u>85 dB(A) Kit</u>
High Idle	102.5	91	87
Torque Converter Stall	102.5-106	93	87.5
High Idle-Lifting Max. Fork Load	100.5	91.5	87.5
Intermediate Gear Loaded to Rated Speed	105.5	(92) <sup>1</sup>	(87) <sup>1</sup>
Full Throttle Acceleration (Maximum Level)	106.5	99	94.5
Cruise in High Gear 2500RPM -		92	91
" " " " 2000RPM -		89	88
" " " " 1500RPM -		85	82.5
Cruise in Intermed. Gear 2700 RPM	-	-	85.5
Cruise in Intermed. Gear 2200 RPM	-	-	85

---

1 - This test is made using the vehicle brakes for loading. A serious brake "squeal" had developed in the vehicle since the original baseline tests. Narrow band analysis showed this noise to be confined to the octave bands centered at 2000 Hz and above. It seemed reasonable to assume the true levels in these bands would not exceed those of Torque Converter Stall condition since the levels in the lower bands were very similar to the TCS levels and generally a bit lower. Thus we "inferred" the A-weighted levels shown by adding the actual octave bands 31.5 to 1000 Hz and the TCS levels at 2000-8000 Hz.

FIGURE 5.3

INITIAL AND FINAL FIELD TEST SOUND  
LEVELS - SPECTATOR NOISE 50 FEET AWAY.

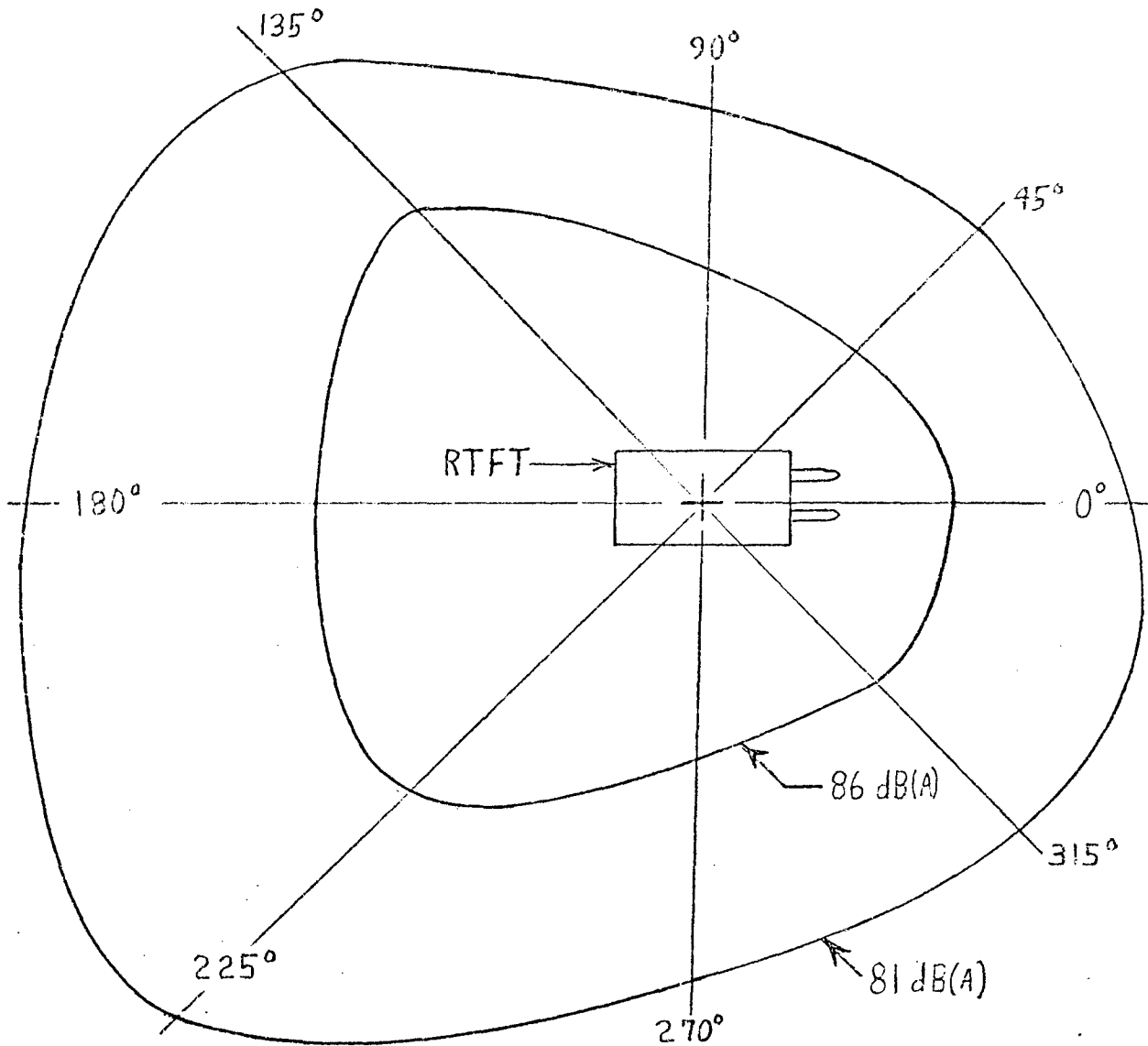
<u>Location/Condition</u>		<u>Original</u>	<u>90 dB(A) Kit</u>
High Idle	Front	79	72.5
	Left	83	76.5
	Rear	83.5	82.5
	Right	82.5	79.5
Torque Converter Stall	Front	82.5	72
	Left	82.5	76.5
	Rear	85.5	82
	Right	86	79
High Idle - Lifting Max. Fork Load	Front	79-82.5	74-78
	Left	83	76.5
	Rear	85	82
	Right	82.5	79
Intermediate Gear Loaded to Rated Speed	Left	88	-
	Right	85	-
Maximum Vehicle Speed No Load	Left	90.5	84
	Right	91.5	85.5

Average reduction in spectator noise: 5 dB(A)

FIGURE 5.4

FINAL ISO-SPL LINES FOR HIGH IDLE CONDITION

Origin of axes at operator's ear which approximates center of vehicle. Level at origin was 91 dB(A).



$\Delta$ dB (A)	DISTANCE FROM OPERATOR EAR ALONG AXIS (FT)							
	0°	45°	90°	135°	180°	225°	270°	315°
-5	23	19	21	36.5	35	38	23	23.5
-10	38	35	35	55.5	61	67	43	42
-15	63	57	68	-*	-*	-*	79	75

\*Distance was greater than 80' radius of paved test area.  
Point could not be taken.

FIGURE VI.

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE FRAME NEAR RIGHT FRONT MOUNT

- As received
- Isolated engine
- △—△ Isolated engine and transmission
- ◇—◇ Lord center bonded mounts at engine rear

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

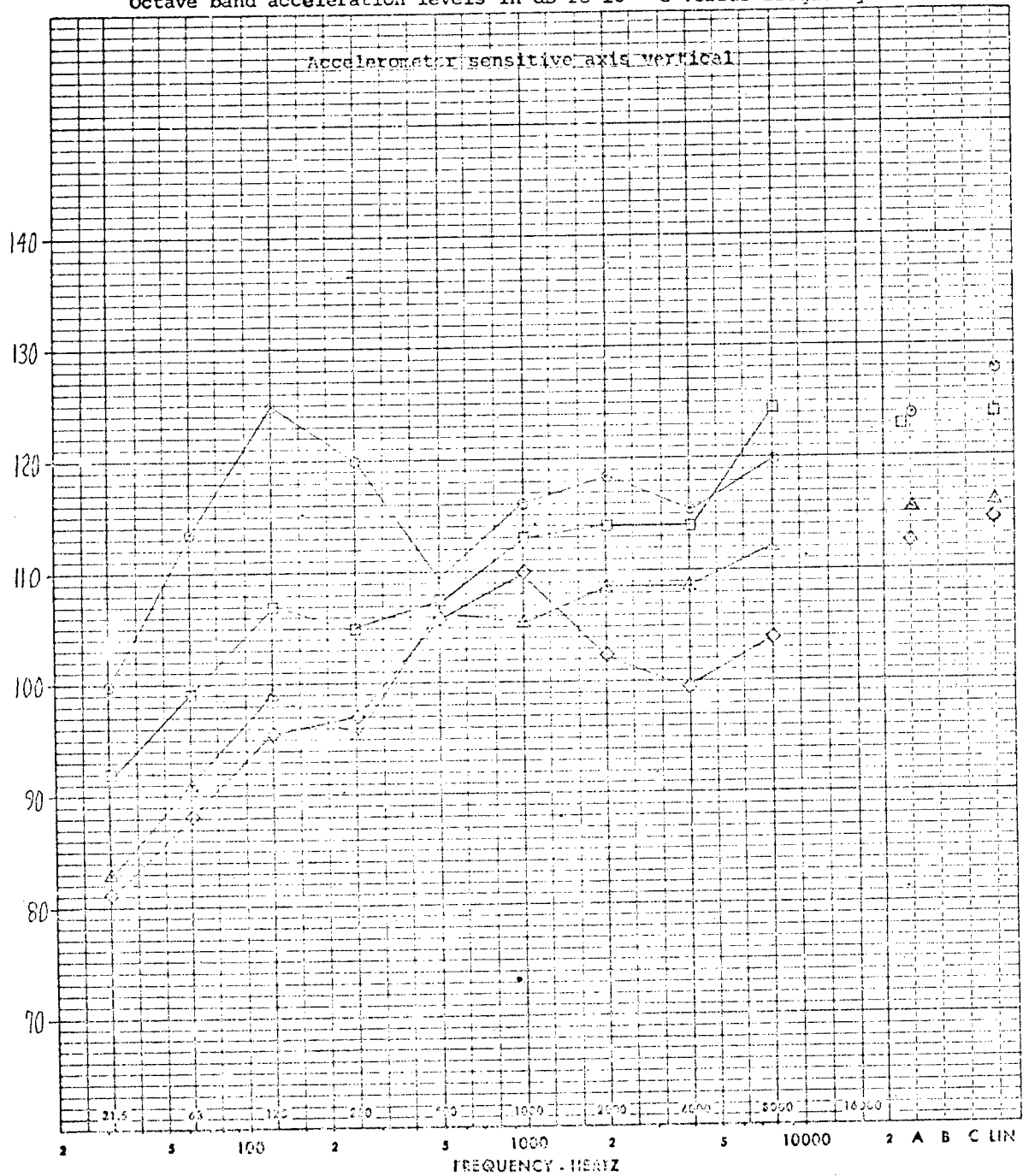


FIGURE V2

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE RIGHT FRONT FENDER

- As received
- Isolated engine
- △—△ Isolated engine and transmission

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

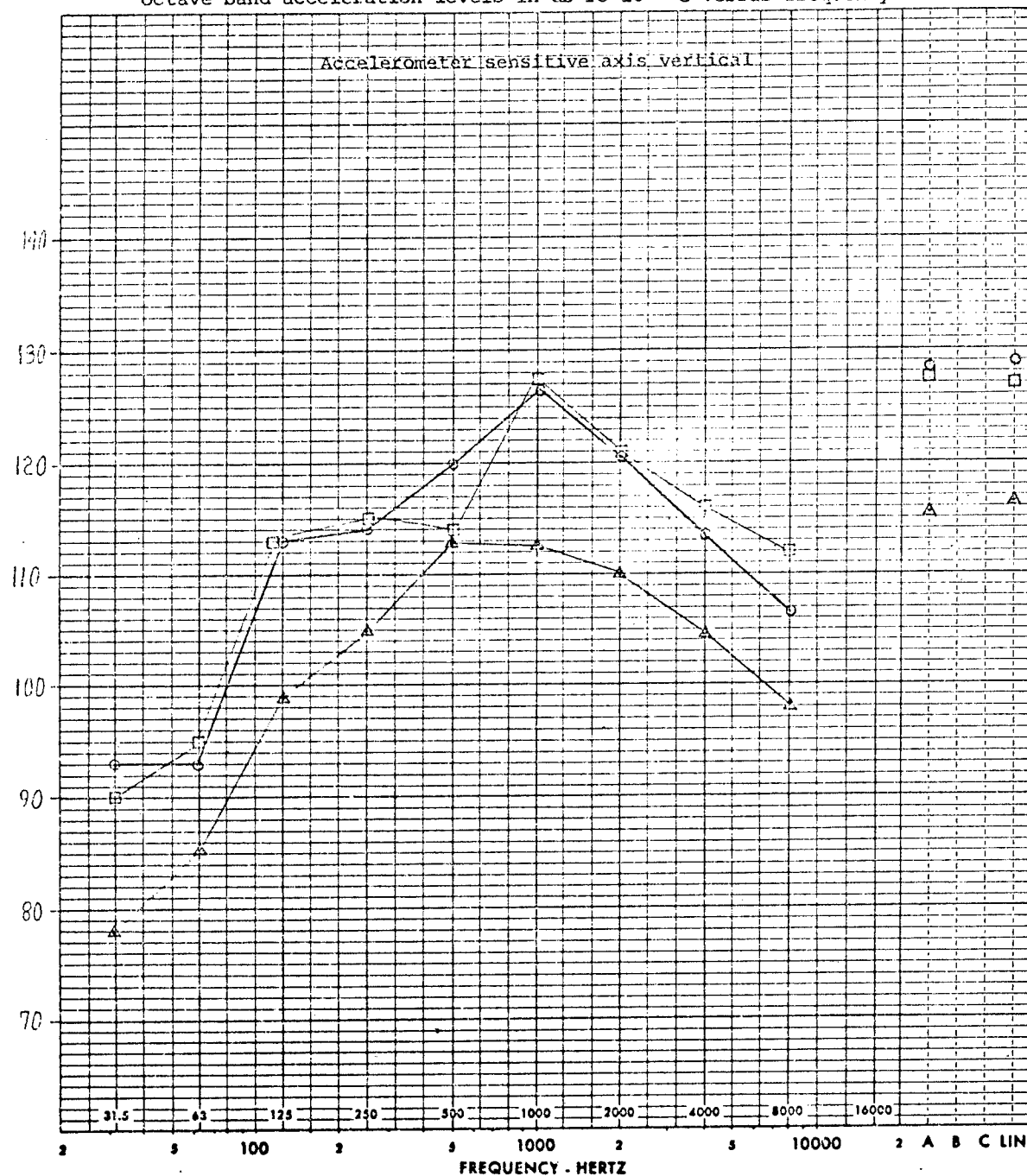


FIGURE V3.1

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE TOP OF RESERVOIR BEHIND OPERATOR

- As received
- Isolated engine
- △—△ Isolated engine and transmission
- ◇—◇ Isolated engine - prop shaft off

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

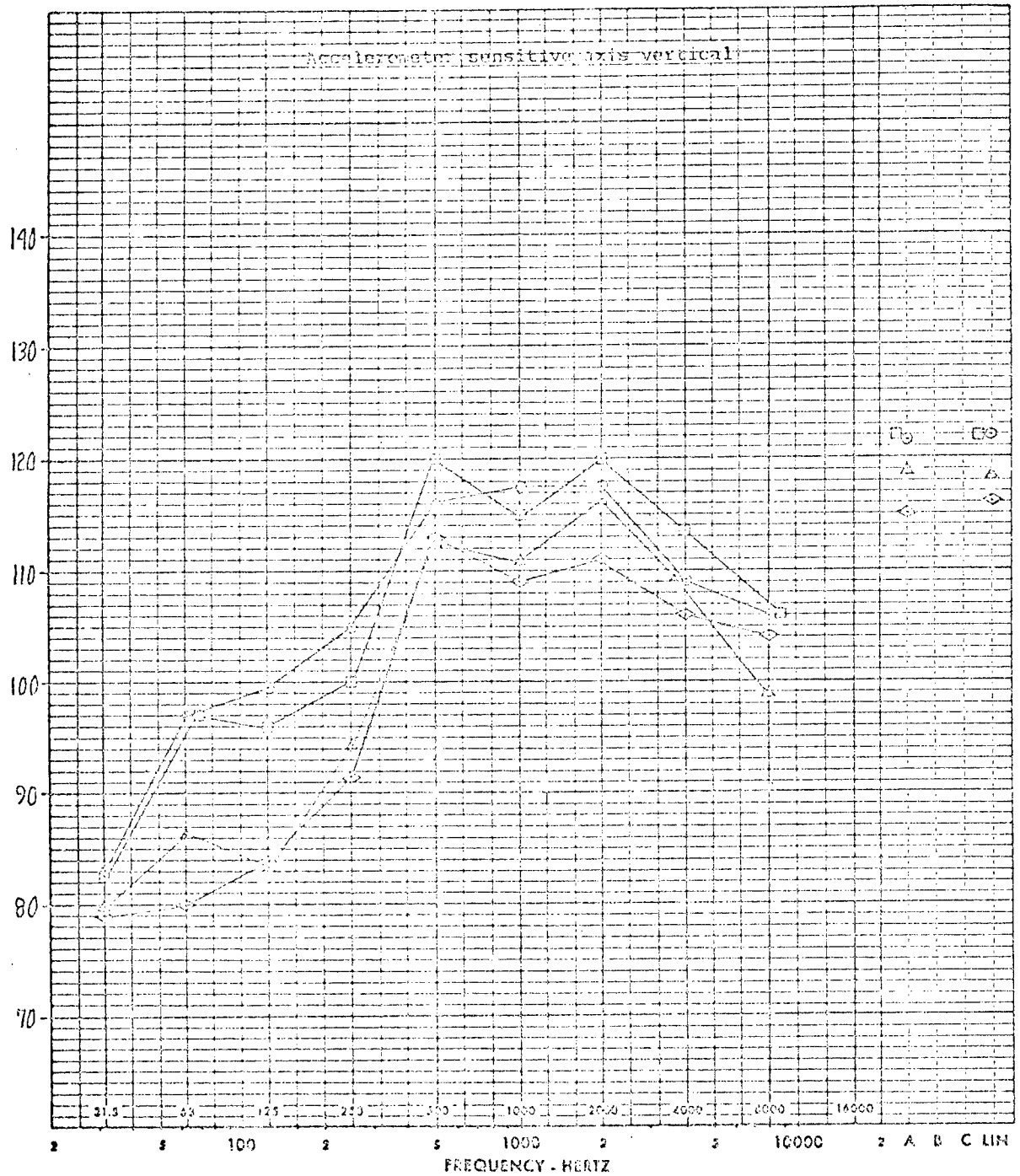


FIGURE V3.2

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE TOP OF RESERVOIR BEHIND OPERATOR

- (1) —△— Isolated engine and transmission
- (2) —■— (1) With hydraulic system removed
- (3) —○— (2) With prop shaft off (T.C. and gears stopped)

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

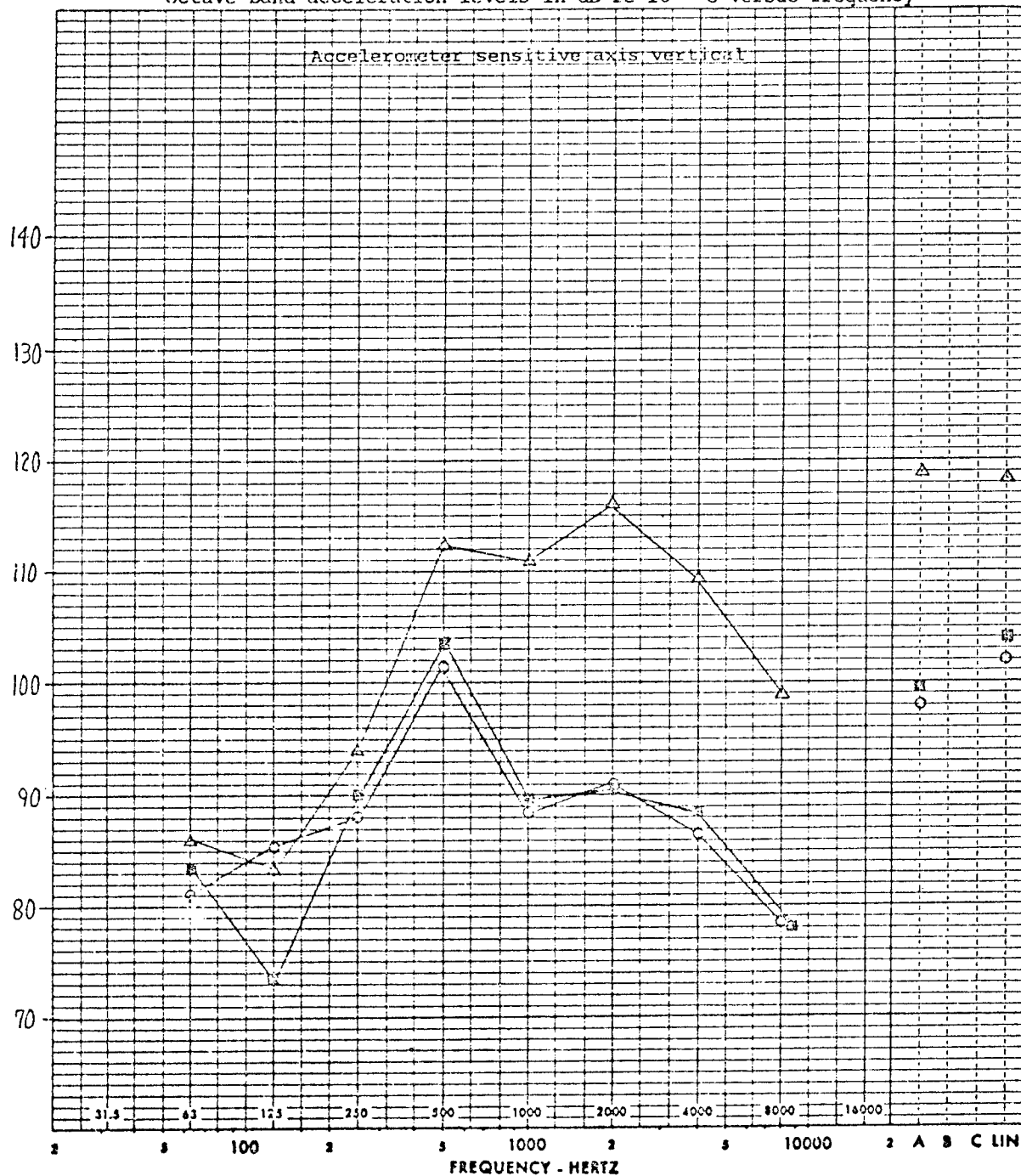




FIGURE V4.1

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE PANEL BY OPERATOR'S RIGHT LEG

- As received
- Isolated engine
- △—△ Isolated engine and transmission
- ◇—◇ Isolated engine -- prop shaft off

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

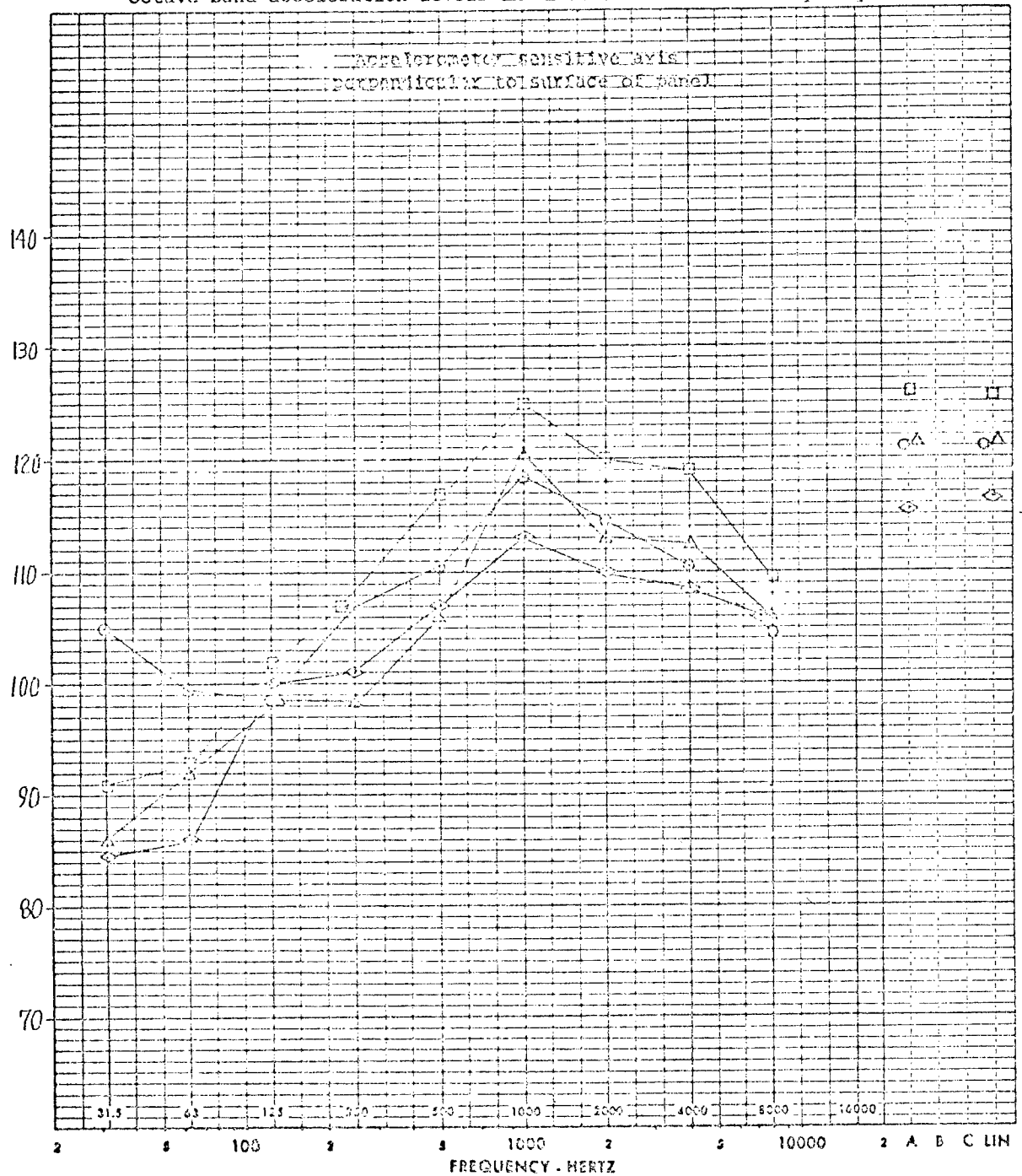


FIGURE V4.2

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE PANEL BY OPERATOR'S RIGHT LEG

- (1) —△— Isolated engine and transmission
- (2) —▲— (1) with panel damping
- (3) —□— (2) with hydraulic system removed
- (4) —○— (3) with prop. shaft off (T.C. and gears stopped)

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

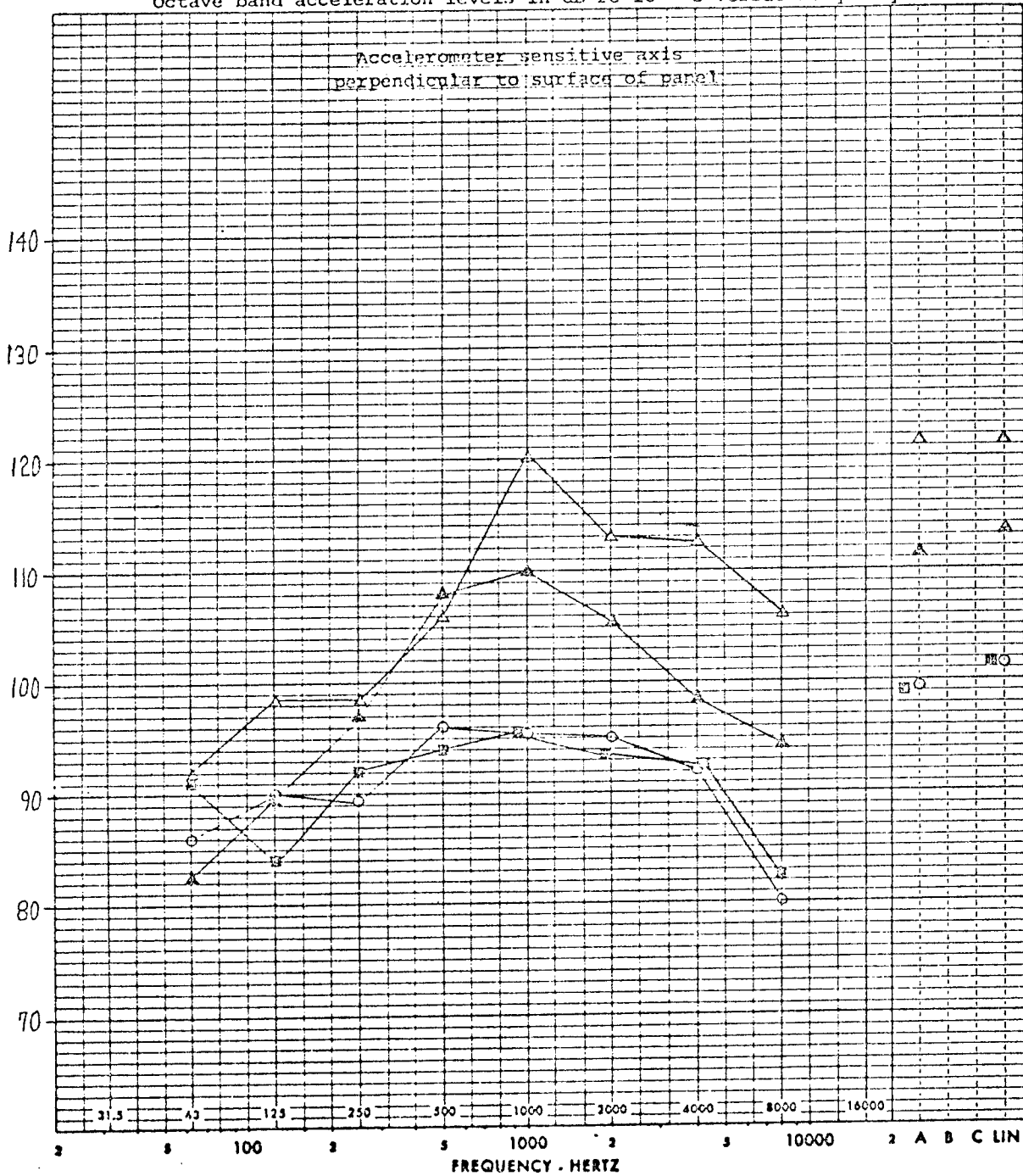


FIGURE V5.1

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE ENGINE HOOD BEHIND BATTERY CASE

- As received
- Isolated engine
- △—△ Isolated engine and transmission
- ◇—◇ Isolated engine - prop shaft off

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

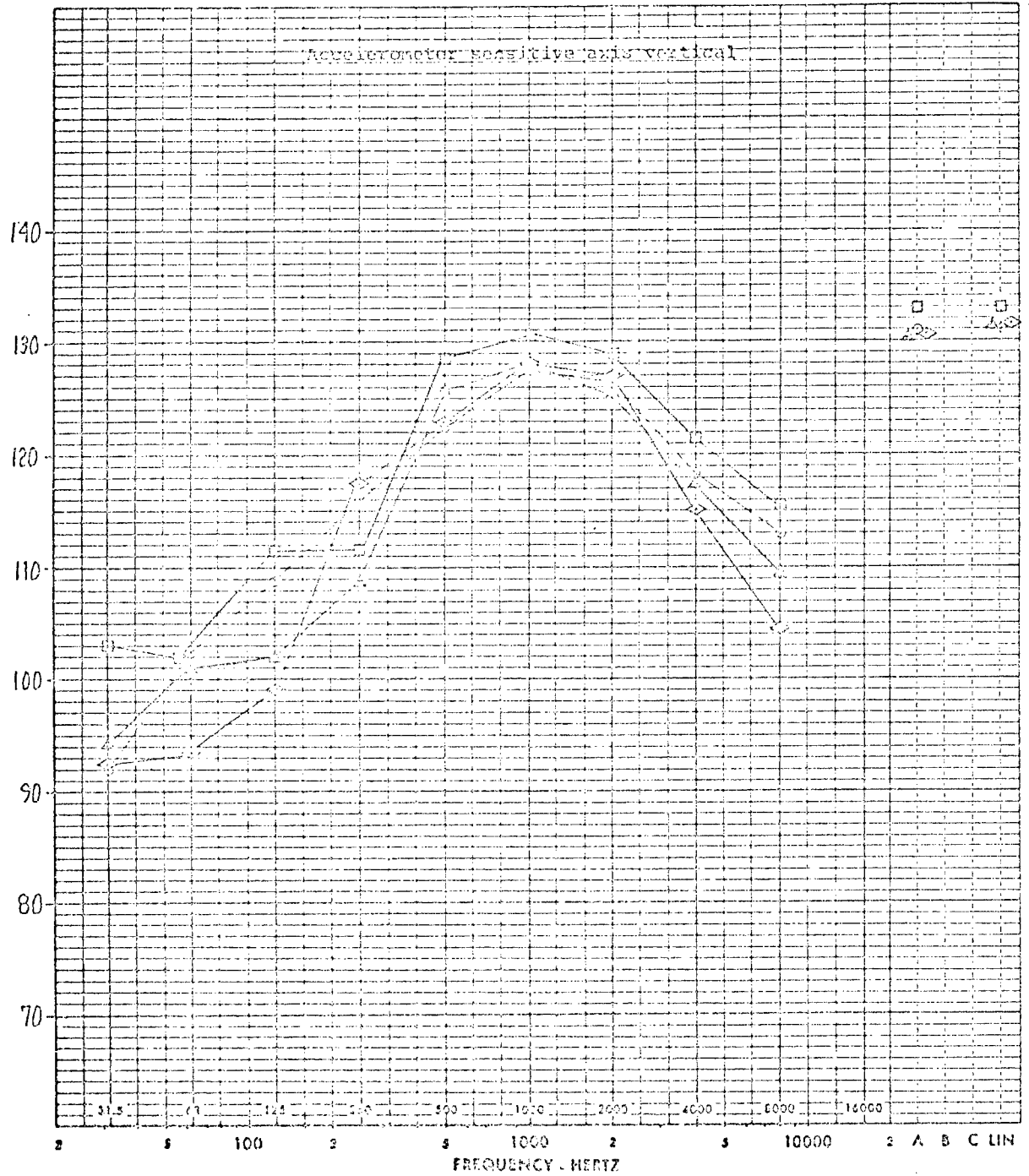


FIGURE V5.2

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE ENGINE HOOD BEHIND BATTERY CASE

- (1) —△— Isolated engine and transmission
- (2) —▲— (1) With panel damping
- (3) —□— (2) With hydraulic system removed

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

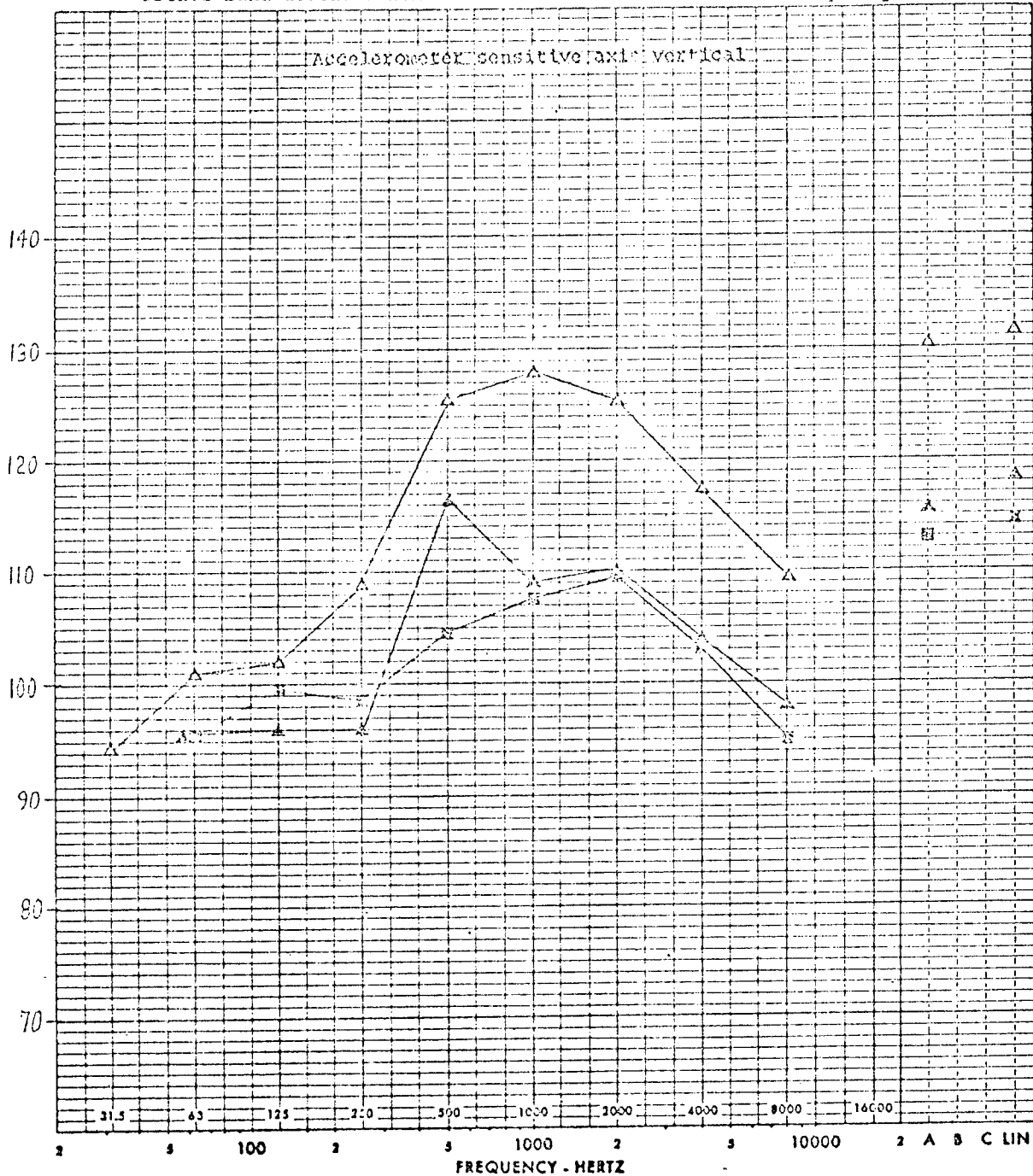


FIGURE V6

H.L. BLACKFORD, INC. TROY, MICHIGAN

## HIGH IDLE VIBRATION LEVELS ON LIFT CONTROL LEVER PANEL

- (1) --△-- Isolated engine and transmission, panel damped  
 (2) --□-- (1) With hydraulic system removed  
 (3) --◇-- (1) With propshaft removed  
 (4) --△-- (3) With damping removed
- Octave band acceleration levels in  $\text{dB re } 10^{-6} \text{ G}$  versus frequency

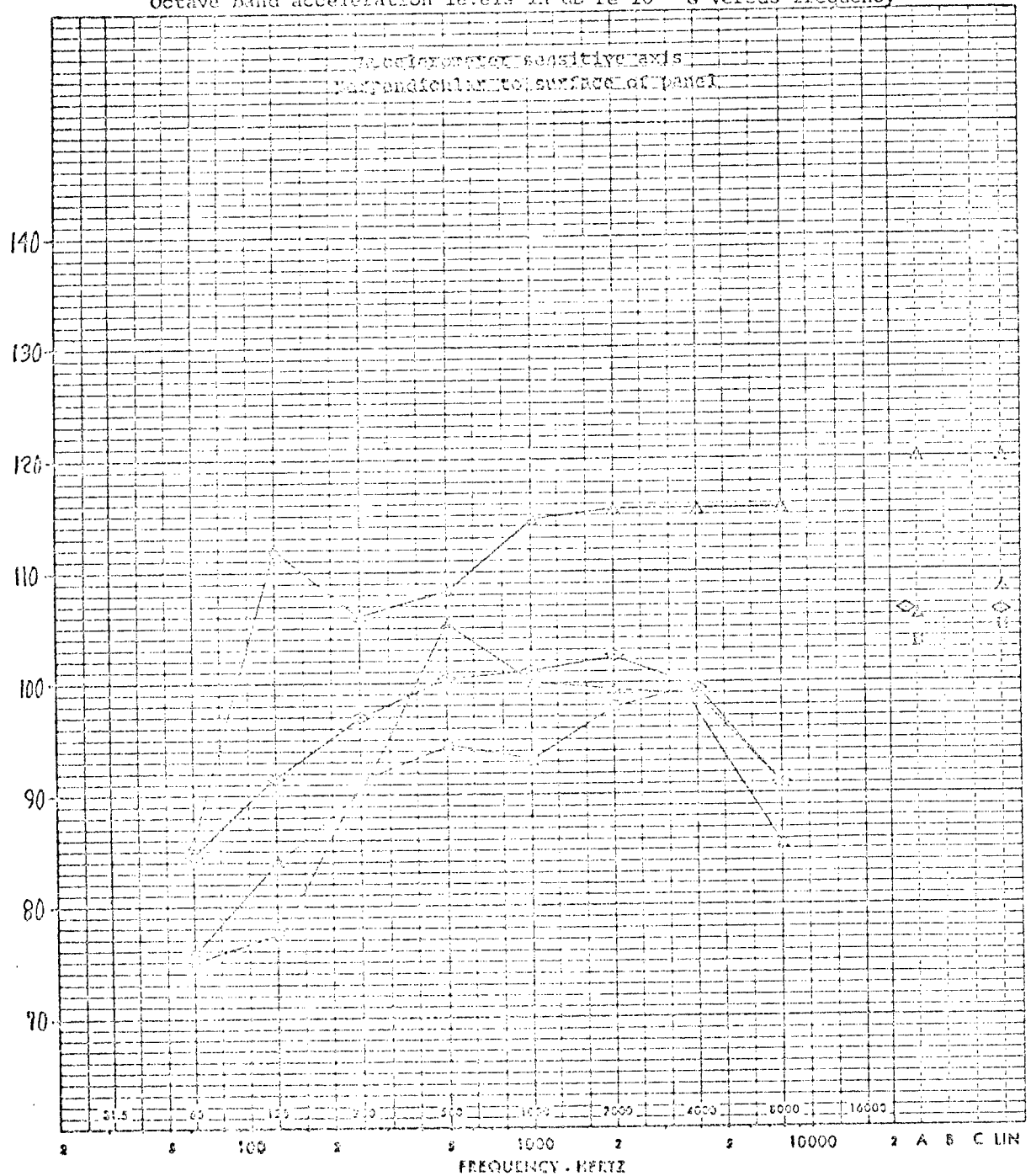


FIGURE V7

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON TOP OF HYDRAULIC RESERVOIR

- (1) —△— Stock condition of reservoir & hydraulic system
- (2) —□— With hydraulic system removed
- (3) —◇— With flexible suction lines
- (4) —▽— (3) plus isolated reservoir

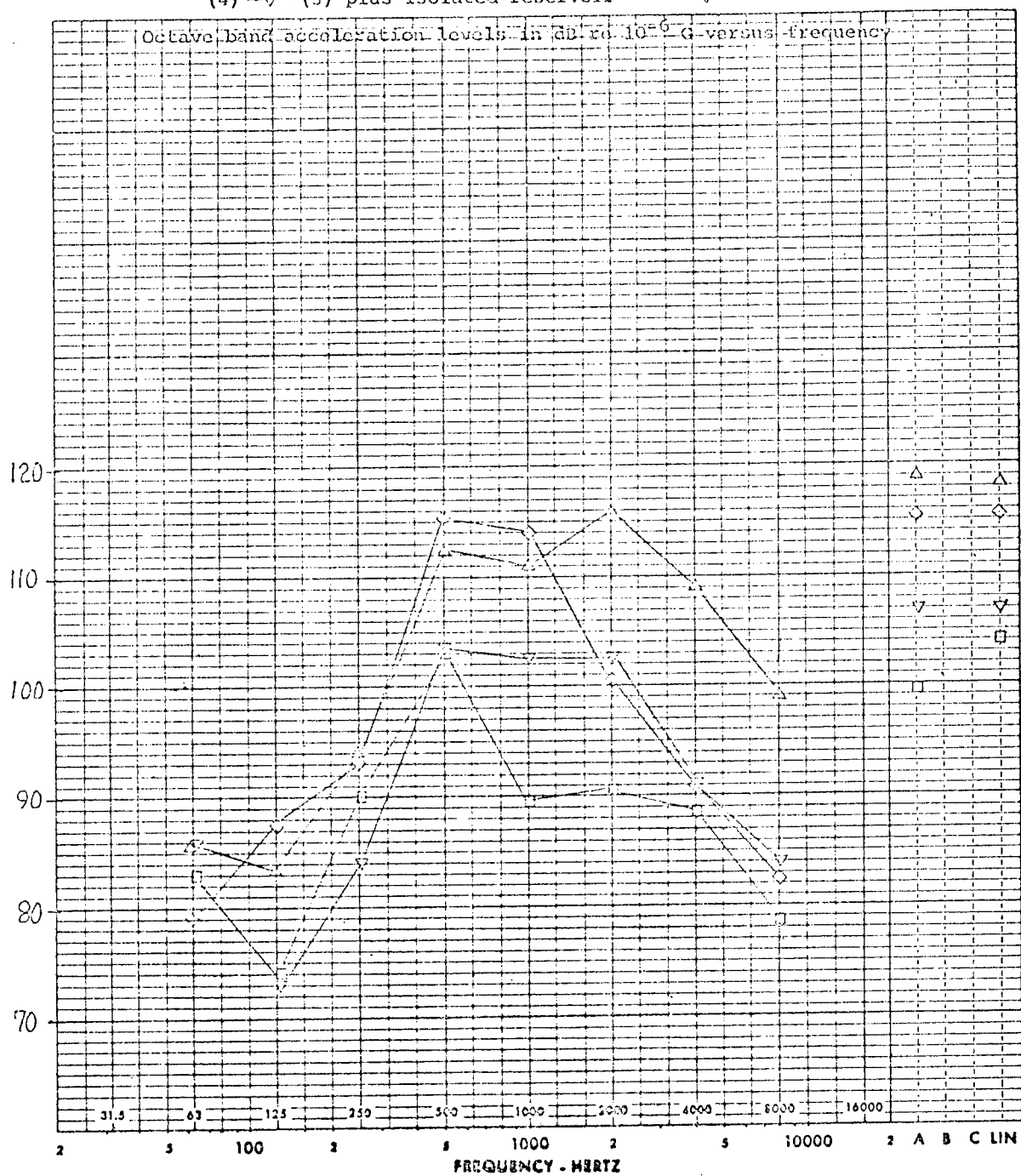


FIGURE V8

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE PANEL BY  
THE OPERATOR'S RIGHT LEG - DAMPED PANEL

- (1)-△-Stock condition of reservoir & hydraulic system
- (2)-□-With hydraulic system removed
- (3)-◇-With flexible suction lines
- (4)-▽-(3) plus isolated reservoir

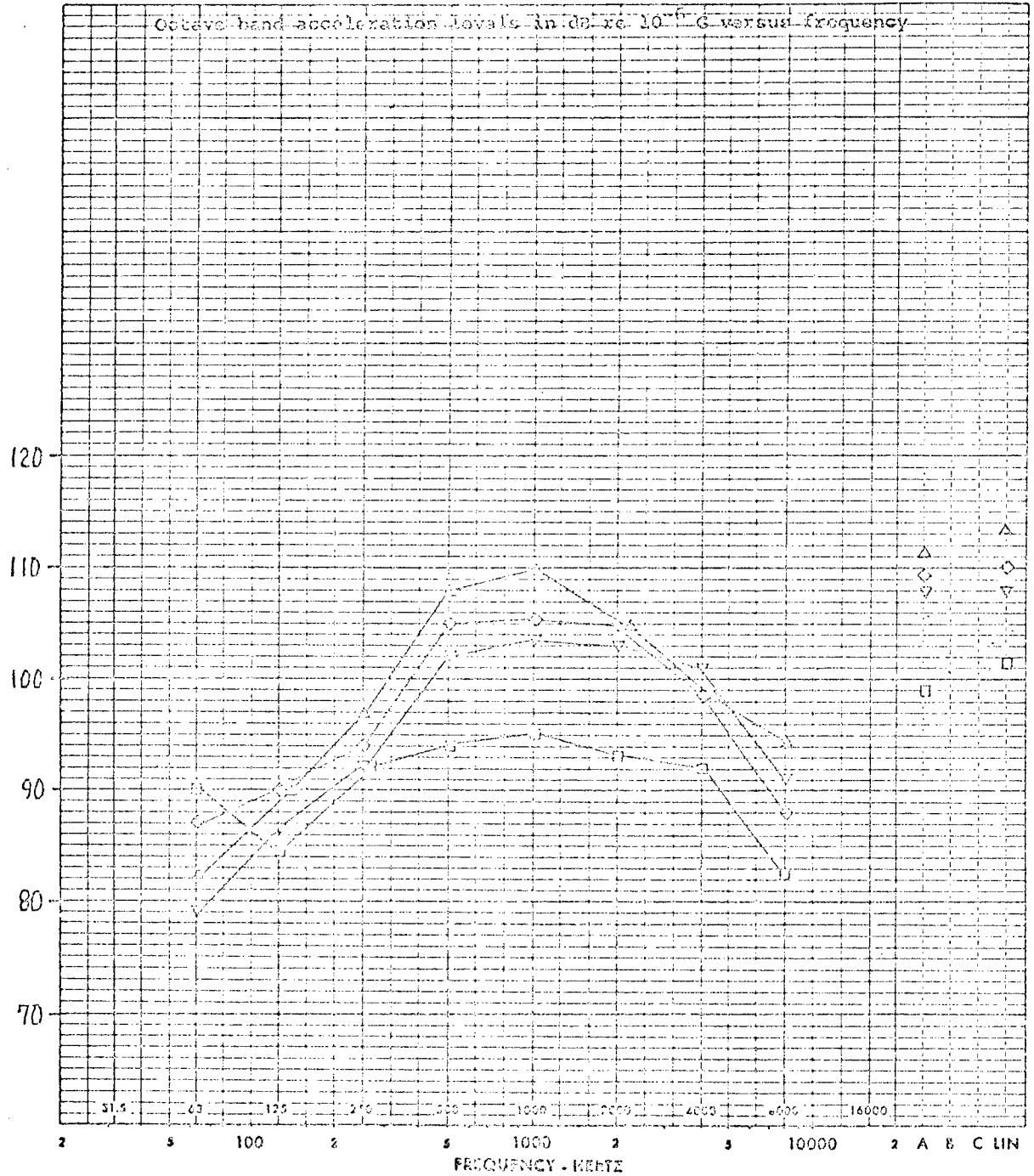


FIGURE V9

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON THE PANEL BY THE OPERATOR'S  
RIGHT LEG STOCK MOUNTED RESERVOIR. EFFECT OF PRESENCE OF  
LIFT CONTROL PANEL ON VIBRATION OF RIGHT SIDE PANEL

- Lift control panel in place.
- ◇—◇ Lift control panel removed.

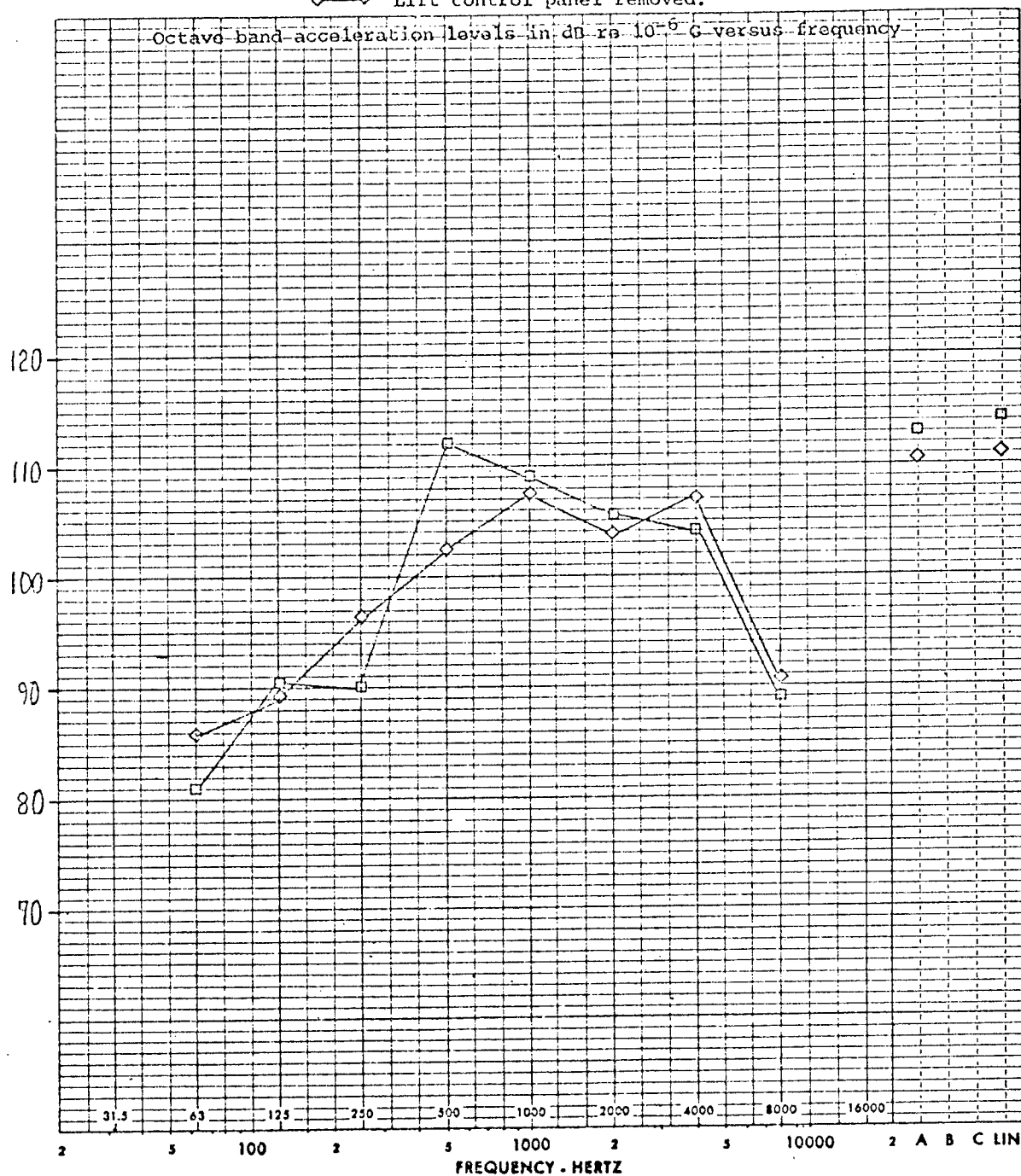




FIGURE V10

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON TOP OF HYDRAULIC RESERVOIR

- No hydraulic pumps
- Commercial Shearing steering pump only
- △—△ Tyrone Steering pump only

Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

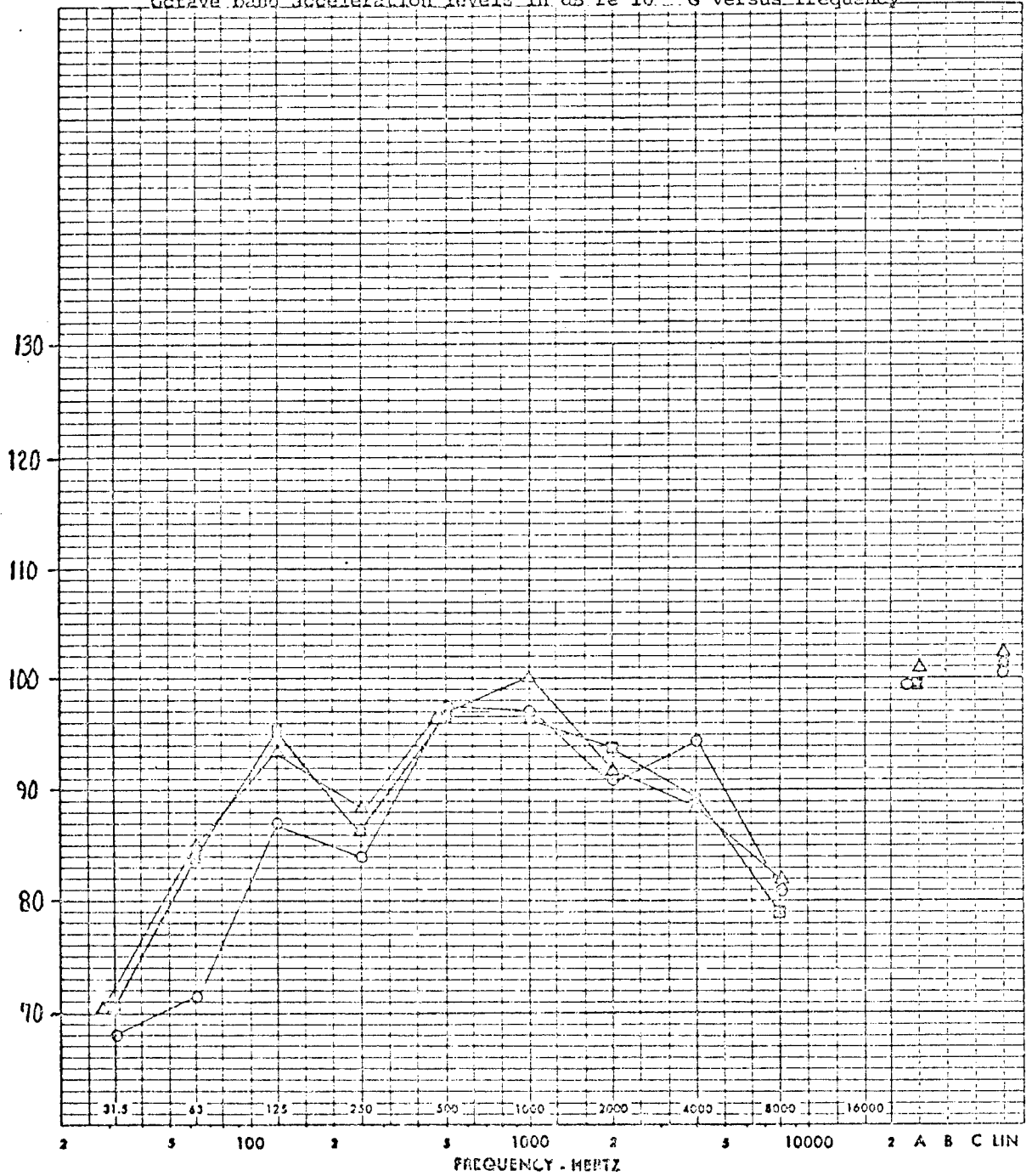


FIGURE V11

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON TOP OF HYDRAULIC RESERVOIR

All three hydraulic pumps running

□ — With Warner-Motive steering pump  
 ○ — With Commercial shearing steering pump  
 △ — With Tyrone steering pump  
 Octave band acceleration levels in dB re  $10^{-6}$  G versus frequency

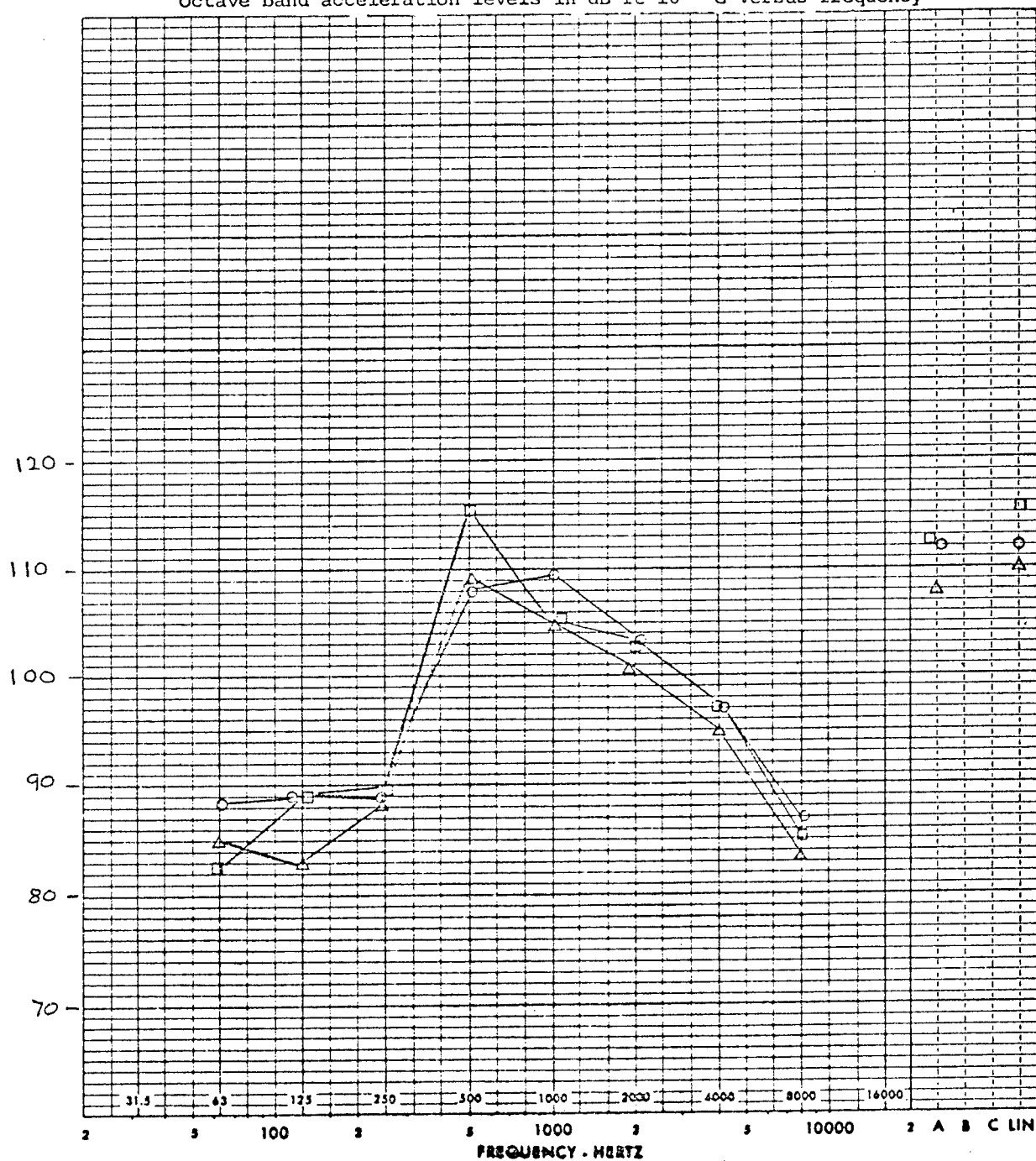


FIGURE V12

H.L. BLACHFORD, INC. TROY, MICHIGAN

HIGH IDLE VIBRATION LEVELS ON TOP OF HYDRAULIC RESERVOIR

- (1)—○— As received
- (2)—△— Tyrone steering pump, isolated reservoir and flexible suction lines
- (3)—□— (2) with Commercial Shearing lift pump

Octave band or weighted acceleration levels in dB re  $10^{-6}g$

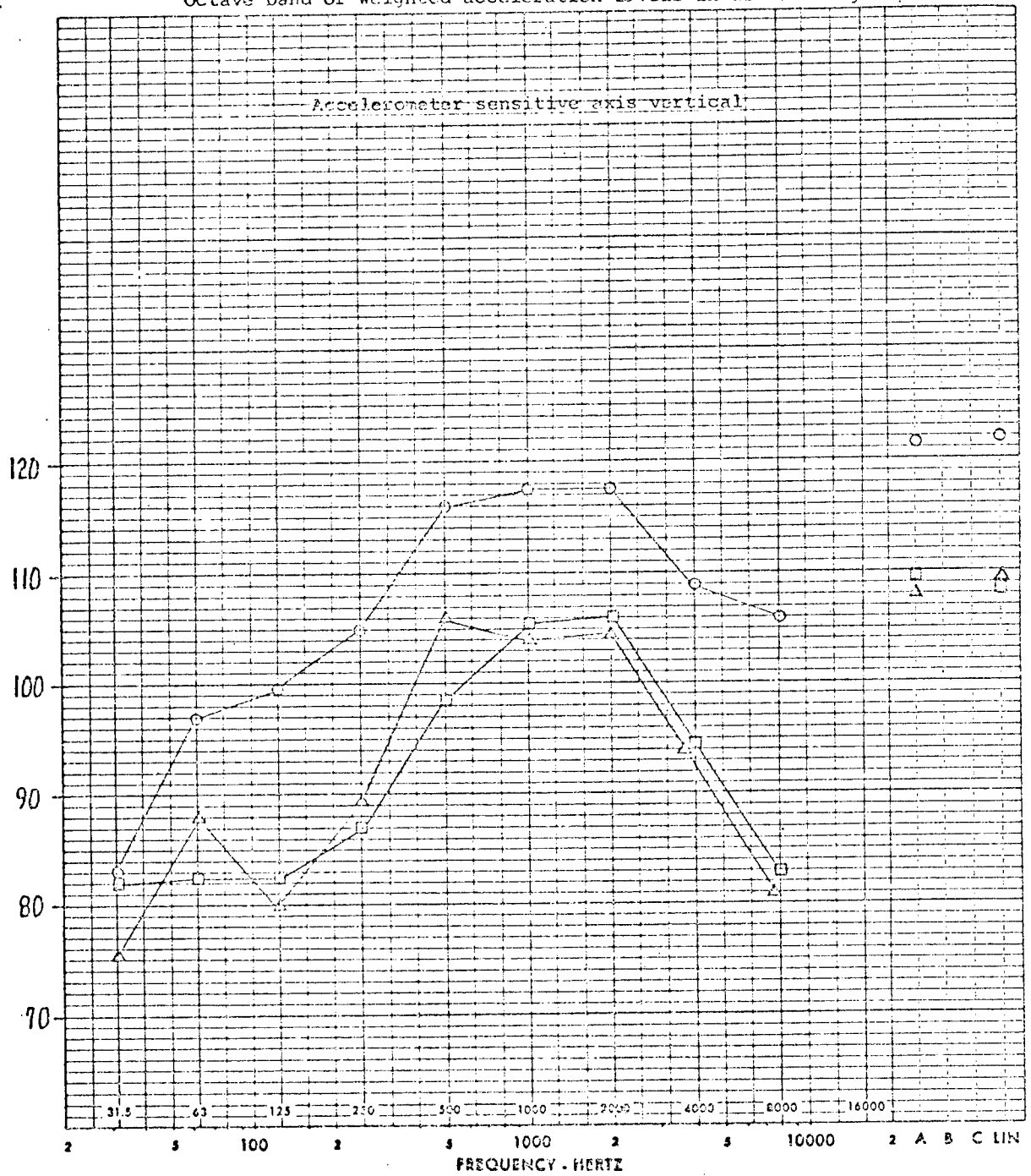
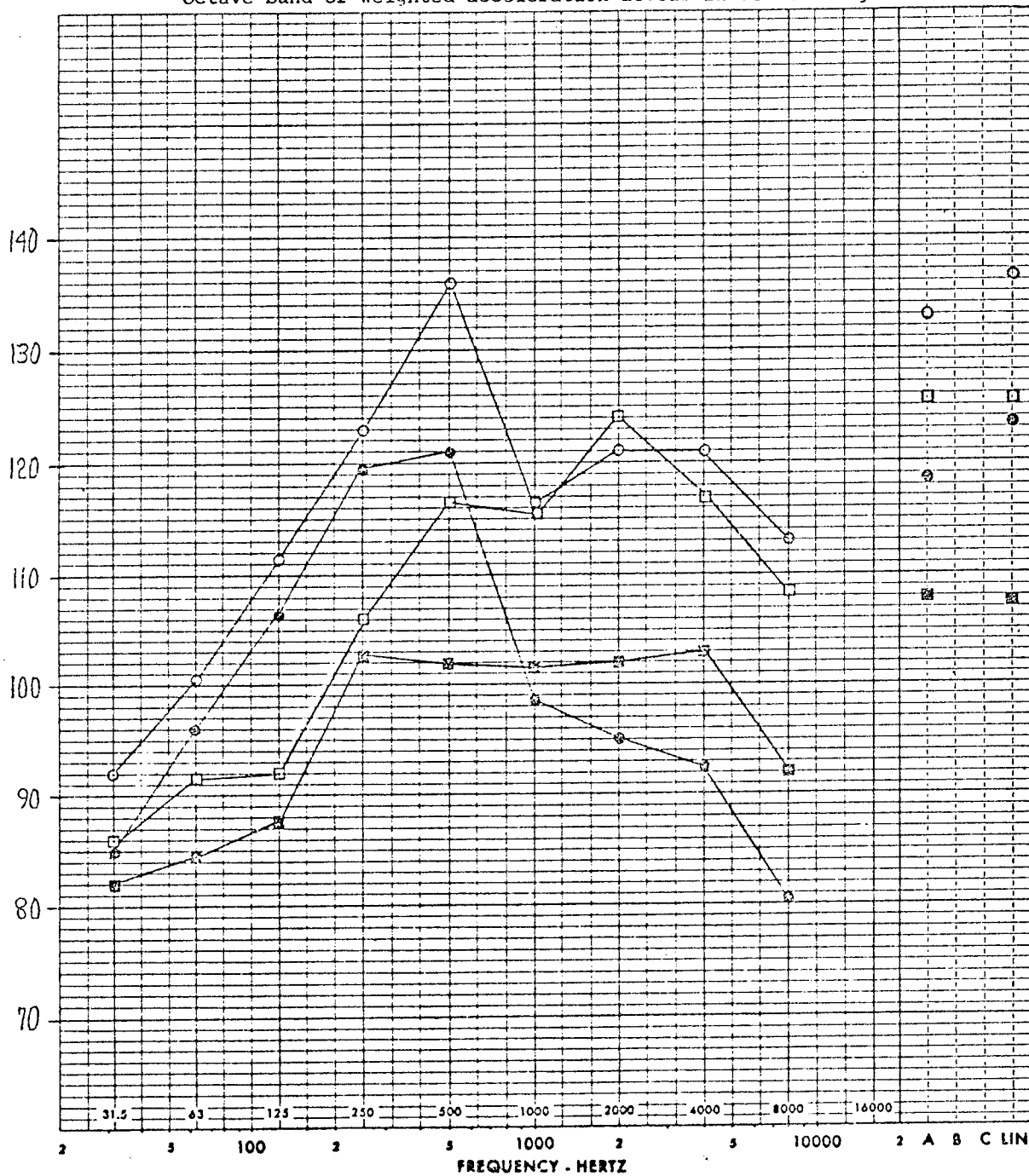


FIGURE V13

H.L. BLACHFORD, INC. TROY, MICHIGAN

VIBRATION LEVELS ON AIR CLEANER-HIGH IDLE ENGINE CONDITION

- End cap - as received
  - Side, as received
  - End cap. Damped and added intake silencer in place
  - Side. Damped and added intake silencer in place
- Octave band or weighted acceleration levels in dB re  $10^{-6}g$



## APPENDIX A

### NOISE, VIBRATION AND OTHER TEST INSTRUMENTATION USED IN REFT PROGRAM

#### Electronic and Test Equipment Listed by Manufacturer

##### Bruel & Kjaer

Level Recorder, Type 2305; Serial No. 144288  
Impulse Sound Level Meter, Type 2204; Serial No. 313730  
Sound Level Meter, Type 2206; Serial No. 338978  
Beat Frequency Oscillator, Type 1022; Serial No. 145909  
Frequency Analyzer, Type 2107; Serial No. 147796  
Band Pass Filter, Type 1612, Serial No. 144100  
Microphone Amplifier, Type 2603; Serial No. 204230  
Pistonphone, Type 4220; Serial No. 147402  
Accelerometer Set, Type 4335; Serial No. 118349  
Microphone Calibrator, Type 4142; Serial No. 401121  
1" Cathode Follower, Type 2613; Serial No. 146873  
1/2" Cathode Follower, Type 2615; Serial No. 468145  
1/2" Condenser Microphone, Type 4133; Serial No. 311214  
1/2" Condenser Microphone, Type 4134; Serial No. 342210  
1/2" Condenser Microphone, Type 4148; Serial No. 260308

##### Hewlett-Packard

Electronic Counter, Type 5211A; Serial No. 548-00569

Ampex Corp.

Tape Recorder, Type 602-2 (Binaural); Serial No. None

Tape Recorder, Type AG-440; Serial No. 2841112

Nagra-Kudelski

Monaural Tape Recorder, Type III N; Serial No. BH-645881

Electro-Voice

(2) Dynamic Microphone, Type 655C; Serial No. 9120

General Radio Corp.

Random Noise Generator, Type 1390-B; Serial No. 8043

Lewis Engineering Co.

Portable Potentionmetric Pyrometer, Type 10003; Serial No. 3

Allied Radio

Oscilloscope, Type KG-2000; Serial No. None

Knight Amplifier, Type KM-15; Serial No. None

Altec Lansing

Amplifier, Type 711A; Serial No. 4482

Speakers (2); Type 841 B; Serial No. None

McIntosh

Amplifier, Type MC-30; Serial No. 8B484

Pre-Amplifier, Type C-8; Serial No. 4B22

Davis

Flow Anemometer

Bacharach

Sling Psychrometer; Code 12-7012

APPENDIX B

COST ESTIMATES

The attached figure provides budgetary quotations on kit parts and lists the vendors supplying quotations. In round figures, we would estimate the 90 dB(A) Kit cost at \$500 and the 85 dB(A) Kit cost at \$720.

Based on our experience with the prototype we estimate man-hours of labor for installation of the kits as follows:

<u>Activity</u>	<u>Man-Hours</u>
Disassembly	15
Rework existing parts	35
Reassembly	25
Adjustment for proper fit	10
Install Shield	<u>5</u>
	90 Hours

These estimates are for average time for technically competent personnel reworking a large number (at least 20) vehicles.



HLBI OR VENDOR* PART NUMBER	NAME OF PART	QTY/ RFT	TOOLING	ea 1000	TOTAL	ea 500 RTFT	TOTAL	QUOTED BY VENDOR NO.
T-13928*	Exhaust Muffler	2	\$ -	\$59.60	\$119.20	\$59.60	\$119.20	1
916097*	Taper Twist Fan	1	9400	11.49	11.49	12.06	12.06	2
5144796*	Fan Spacer (.80")	2	-	2.25	4.50	2.25	4.50	3
5109638*	Crank Pulley (5.38" Dia.)	1	-	15.67	15.67	15.67	15.67	3
14700	Seal Plate, Shroud	2	352	2.87	5.74	2.93	5.86	4
13501	Shroud, Orifice-Weldmount	1	-	16.20	16.20	20.60	20.60	4
13502	Shroud, Orifice-Weldmount	1	660	16.20	16.20	20.60	20.60	4
13600	Trans. Iso. Pad, Front	4	-	.74	2.96	.80	3.20	5
13700	Engine Iso. Pad, Front	2	-	2.34	4.68	2.59	5.18	5
13800	Engine Iso. Pad, Side	4	-	1.14	4.56	1.25	5.00	5
14000	Trans. Iso. Pad, Rear	4	-	.29	1.16	.31	1.24	5
16300	Isolator, Valve, Lower	4	-	.56	2.24	.50	2.40	5
16401	Isolator Pad	16	-	.18	2.88	.20	3.20	5
16402	Isolator Pad	14	-	.13	2.52	.20	2.80	5
16500	Isolator Pad, Reservoir	2	-	.42	.84	.44	.88	5
16601	Tube, Lateral Isolation	10	-	.058	.58	.058	.58	6
16602	" " "	2	-	.083	.17	.083	.17	6
16603	" " "	4	-	.070	.28	.070	.28	6
16604	" " "	8	-	.021	.17	.021	.17	6
16605	" " "	4	-	.022	.09	.022	.09	6
14600	Load Plate, Engine Front, ISO	1	160	2.19	2.19	2.25	2.25	4
12600	Cover, Boom Center	1	680	21.35	21.35	23.85	23.85	7
14900	Retainer, Flexible Cover	1	256	3.01	3.01	3.20	3.20	4
16000	Flex, Cover-Boom	1	180	13.98	13.98	14.12	14.12	7
13900	Floor Mat	1	210	23.70	23.70	23.92	23.92	7
12800	Cover, Panel Opening	1	192	1.02	1.02	2.06	2.06	4
13100	Cover, Boom Pivot	1	680	3.00	3.00	3.18	3.00	4
14300	Retainer, Duct Liner	1	400	10.60	10.60	13.10	13.10	4
14800	Retainer, Reservoir Seal	1	216	3.05	3.05	3.40	3.40	4
17500	Hood Plug, Exhaust Pipe	2	250	1.61	3.22	1.80	3.60	4

HLBI OR VENDOR* PART NUMBER	NAME OF PART	QTY/ RFT	TOOLING	ea 1000	TOTAL	ea 500 RFT	TOTAL	QUOTED BY VENDOR NO.
17600	Plug, Battery Box Rear	1	144	1.20	1.20	1.40	1.40	4
12900	Cover, Left Side Panel	1	120	1.49	1.49	1.55	1.55	4
13200	Cover, Hood, Left Side	1	192	1.66	1.66	1.85	1.85	4
16100	Seal Plate	1	216	1.16	1.16	1.22	1.22	4
16700	Seal Plate, Upper	1	292	1.36	1.36	1.42	1.36	4
16800	Seal Plate, Lower	1	292	1.22	1.22	1.42	1.36	4
16900	Cover, Fitting - Assy	1	320	1.60	1.60	1.82	1.82	4
17400	Duct Liner	1	430	6.71	6.71	6.74	6.74	4
15500	Absorption, Right Rear-1	1	470	2.47	2.47	2.49	2.49	7
15400	Absorption, Right Front	1	540	4.60	4.60	4.64	4.64	7
15700	Absorption, Right Rear-2	1	405	2.28	2.28	2.31	2.31	7
15200	Absorption, Hood Front	1	510	3.32	3.32	3.34	3.34	7
17000	Absorption, Hood Center-2	1	570	3.39	3.39	3.43	3.43	7
15100	Absorption, Hood Center	1	585	4.73	4.73	4.77	4.77	7
15300	Absorption, Left Side	1	675	8.98	8.98	9.07	9.07	7
15000	Absorption, Hood Rear	1	600	3.72	3.72	3.76	3.76	7
P2-150-2D4-D*	Pump (PC-114300) (Tyrone)	1	-	75.50	75.50	77.50	77.50	8
	Spline Adaptor		-					3
17801	Hose-Suction	1	-	5.60	5.60	5.60	5.60	9
17802	Hose-Suction	1	-	7.50	7.50	7.50	7.50	9
5BM-0072*	Silencer, Intake	1	-	30.60	30.60	30.60	30.60	10
15900	Air Cleaner MTG Bracket	2	336	2.65	5.30	3.00	6.00	4
17100	Operator Shield	1	-	205.20	205.20	228.40	228.40	11
13100	Reservoir Seal	1	165	1.50	1.50	1.50	1.50	7, 12
13200	Retaining Cap	38	-	.03	1.14	.03	1.14	12
TOTAL				\$ 20,333	\$ 679.48		\$ 725.29	

VENDORS SUPPLYING BUDGETARY QUOTES

<u>Vendor No.</u>	<u>Name and Address</u>
1	Nelson Muffler P.O. Box 308, Stoughton, Wisconsin 53589. (Mr. Dale Zuhse)
2	Schwitzer Div. of Wallace-Murray Corp. PO Box 80-B, Indianapolis, Indiana
3	Detroit Diesel Allison Div. of General Motors 13400 West Outer Drive, Detroit, MI 48228
4	Sons Tool & Engineering Co., Inc. 2262 Terminal Rd., St. Paul, Minn. 55113 (Mr. Jacobs or Mr. Swanson)
5	Fabrecka Products Co. 1190 Adams St. Boston, Mass. 02124 (Mr. McLaughlin)
6	Rubber Mat'ls Corp. 1263 Souter Blvd., Troy, MI 48084
7	H.L. Blachford, Inc., 1855 Stephenson Hwy., Troy, MI 48084
8	Tyrone Hydraulics Inc., PO Box 511 Corinth, Miss. 39834 (Mr. Geo. Brooks)
9	Weatherhead tubing. (Obtained from The Chas. A. Strelinger Co., Warren, MI; but available generally)
10	Donaldson Muffler Co. 1400 West 94th Street Minneapolis, Minnesota 55431
11*	D&M Truck Top, 8525 Puritan Ave., Detroit, MI (Mr. Aaron Greenspon)
12	R. Baker estimate
13**	Ross Sheet Metal Co., 2300 Hilton Road Ferndale, MI 48220 (Mr. D. J. Ross)

\* This vendor was very responsive in providing prototype but is not "geared" to volumes of 500 and 1000 at present. Other quotes should probably be obtained if large no. of parts are contemplated.

\*\* This vendor also not accustomed to large quantity but provided sheet metal prototype parts and was willing to quote 500 & 1000. He would not "tool" parts. Might be suitable for small quantities.

## APPENDIX C

### DANA CORPORATION INSTRUMENTATION

The following page describes the features of the Dana Towing Dynamometer used in the engine cooling and drawbar pull tests. Other instrumentation was as follows:

Temperature: Doric PS-300 Thermocouple Indicator Serial No. 11631.

Shaft Speeds: Hewlett Packard 5210 Electronic Counter Serial No. 120-00655. Hewlett Packard AC Amplifier 466A, Serial No. 226-02742. Servo-Tek 0-3000 RPM Meter and Generator.

Road Speed: Labeco DDI MPH Indicator, Serial No. 11631.

Exhaust back pressure: Marshalltown 0-5 psi Pressure Gage.

### DANA TOWING DYNAMOMETER

The Dana Corporation Towing Dynamometer is presently designed to simulate steady state grade loads to the towing vehicle. Performance specifications of unit are as follows:

1. Maximum net HP absorption 477 HP @ 13.4 MPH - low axle range  
477 HP @ 18.2 MPH - high axle range
2. Vehicle wt. variable from 26,000 lbs. to 50,000 lbs. with removable ballast.
3. Maximum draw bar pull available 14,000 lbs. @ 2000 psi system back pressure.
4. Maximum grade simulation 25%.
5. Vehicle can be controlled remotely because control console is portable.
6. Dynamometer can be connected directly to tractor via a special plate fastened to fifth wheel plate.

### BASIC OPERATION:

The Dana variable volume axial piston pumps connected to the axles provide the load by restricting the oil flow via a load valve thus creating system back pressure up to 2000 psi max. Heat created is dissipated through an air over oil Hayden cooler with two 48" diameter fans. The load readout device is located in the nose of the converter dolly, which is a 20,000 lb. Lebow load cell.

The load cell produces an 0 to 2 V DC or 0 to 20,000 lbs. signal which is fed into the analog computer which is located in the portable control council. This signal is compared to the desired signal set on the computer by the operator. If the signal received is not in accordance with the signal or load set by the operator, the computer sends a signal to the pumps and load valve to either increase or decrease their output to correspond with the desired signal.

The computer features a feedback system which insures a continuous signal regardless of pavement conditions or elevation changes on the test track curves.

## APPENDIX D

### LOGARITHMIC SUBTRACTION

Levels expressed in decibels are ratios of the actual measured level to some reference level. They cannot be added or subtracted directly by algebraic techniques. Addition can be accomplished by conversion from dB levels back to the actual physical units involved, algebraic manipulation, and re-conversion of the sum (difference) to a new level in dB. This is tedious and so numerous charts and nomograms exist to allow addition or subtraction of two dB levels at a time. One such chart is presented on the following page as Figure D.1.

As a sample problem in calculation of source strength consider Figure 2.8, Effect of Engine Vibration. The level with a hard engine mounting is 87.5 dB(A). With an isolated engine it is 84 dB(A). To find the "source strength" of engine vibration first observe that it plus other sources are 3.5 dB greater than all the other sources at work in Figure 2.8. Enter the chart at 3.5 dB on the abscissa, labeled Numerical Difference between Total and Smaller Levels. Read up to the curved line and over to the ordinate, Numerical Difference between Total and Larger Levels. The difference is 2.5 dB. Therefore engine vibration contributes  $87.5 - 2.5$  or 85 dB(A).

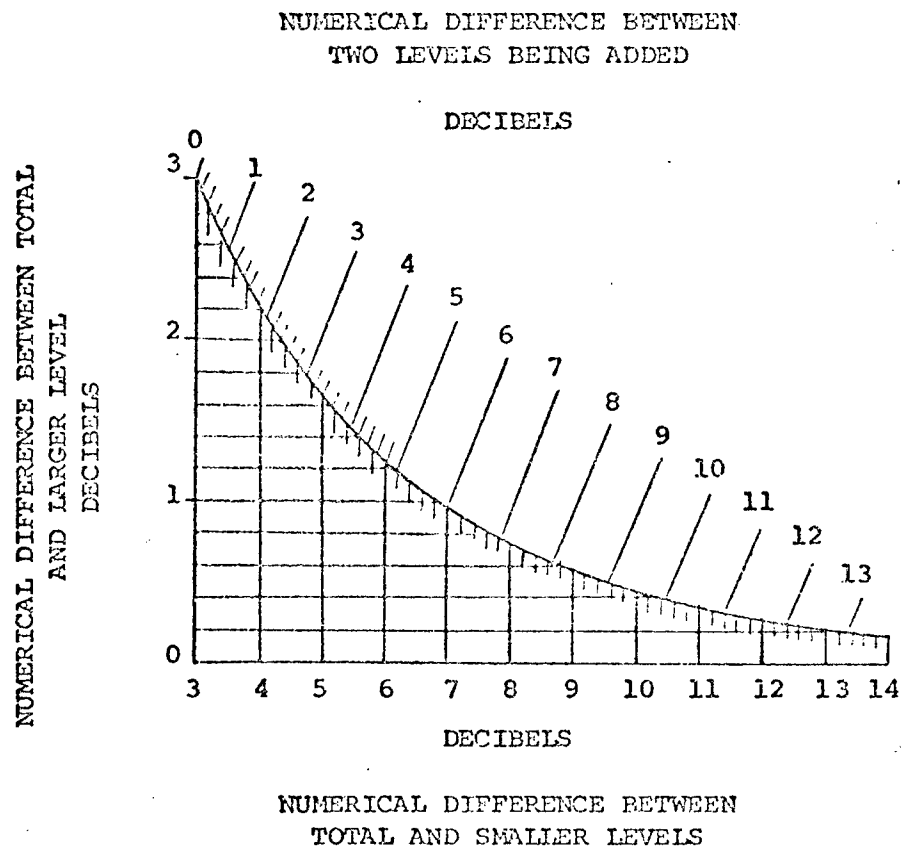


FIGURE D.1

CHART FOR DIRECT ADDITION OR SUBTRACTION  
OF TWO DECIBEL LEVELS

APPENDIX E

TEST COMPONENT INFORMATION



### Air Cleaners and Silencers

#### Production Air Cleaner:

The production air cleaner was manufactured by the Donaldson Company, Inc. The part number (P/N) was FWG 14-0027. It uses a replaceable dry paper filter element P/N P10 4968.

#### Farr Air Cleaner:

The Farr Company Dynacell F 50313 air cleaner was evaluated.

#### Donaldson Intake Silencer:

Model SBM07-0072 Intake Duct Silencer was used.

#### Nelson Intake Silencers:

These were custom designs carrying Nelson Muffler Corporation P/Ns T13989 and T13990.

#### Universal Silencer:

Model SVH-5 Silencer, P/N 15-152 from Universal Silencer Corporation was tested.

### Engine Cooling Fan Information

#### Production Fan:

The production fan was a 6 blade propeller fan with 26 inch blade diameter and 2.75 inch blade pitch. It's Detroit Diesel part number (P/N) is 5171229. It is supplied by both Schwitzer Division of Wallace Murray Corporation and Hayes-Albion Corporation. This fan was a "pusher" delivering air through the radiator from within the engine compartment.

#### Hayes Albion Fan:

This fan was also a 6 blade propeller "pusher" fan. It had a 24 inch blade diameter with a one inch ring shroud attached to the blade tips. Blade pitch was 2.75 inches. The Hayes-Albion P/N was BL5-102F.

#### Schwitzer Fan:

This fan was a 6 blade, 26" diameter, 4.27 inch blade pitch propeller fan. It was not a pusher, but rather drew air through the radiator into the engine compartment. Schwitzer P/N was 916097.

#### Shrouds

Both of the alternative fans were fitted with simple orifice shrouds with 1/2 inch clearance to blade tip or ring. Fans were immersed in shrouds to the dimensions recommended by the manufacturer.

#### Laboratory Cooling Comparison

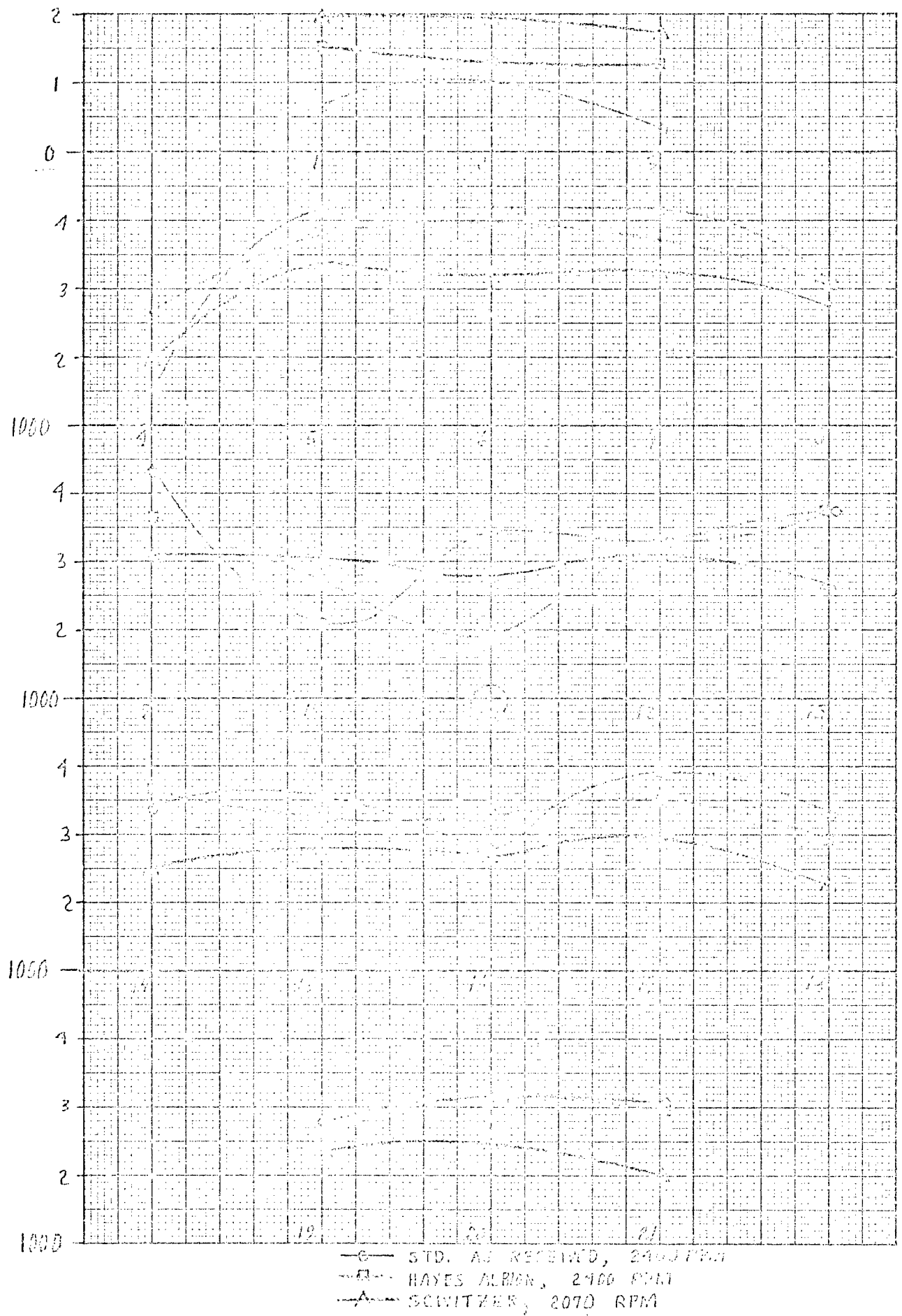
An array of 21 points on the radiator screen was established for flow measurements with a propeller anemometer. This took about 25 minutes for each fan tested. Initial and final ambient and top tank temperatures were recorded. The vehicle was warmed up at High Idle prior to testing and held at High Idle during testing.

Figure E.1 shows the velocity profiles for each fan. Air flows and  $\Delta T$  (Top tank minus ambient) temperature differences are compared below:

<u>Fan</u>	<u>Speed (RPM)</u>	<u>Air Flow (CFM)</u>	<u>Orig. <math>\Delta T</math> (F<sup>o</sup>)</u>	<u>Final <math>\Delta T</math> (F<sup>o</sup>)</u>
Production	2400	16,000	100	102
Hayes Albion	2400	16,100	88	96
Schwitzer	2070	14,300	109	90

FIGURE E.1

# RADIATOR SCREEN VELOCITY PROFILES



## Hydraulic Pumps

### Production Steering Pump:

Warner Motive Division of Borg-Warner Corporation P/N MHD3-23EH5-1-R01. 30 gallons per minute (gpm) at 2000 p.s.i. and 3000 RPM, 13 gpm at 2000 p.s.i. and 1200 RPM. Nine tooth gear pump.

### Production Lift Pump:

Warner Motive P/N MHD3-3-28-EZ5-1R01. Dual Chamber. 35 and 15 gpm respectively at 2000 p.s.i. and 3000 RPM. 9 tooth gear pump.

### Tyrone Steering Pump:

Tyrone Hydraulics Inc. P/N P-150-2D4-D (PC-114300). 15.0 gpm at 1200 RPM and 2000 p.s.i. 10 tooth gear pump.

### Commercial Shearing Steering Pump:

Commercial Shearing, Inc. P/N P30A-194 ECE 2AC1099. 11.5 gpm at 2000 p.s.i. and 1200 RPM.

### Commercial Shearing Lift Pump:

P/N P30B 178-GY OJ15-25-CAB07-1 Tandem 8.0 and 15.0 gpm respectively at 1200 RPM and 2000 p.s.i.

## Exhaust Muffler Information

1. Stemco Manufacturing Company, Inc. mufflers P/N 9207. 8.5 inch dia. by 36 inches long. Double wrapped.
2. Riker Manufacturing, Inc. mufflers P/N 9XD354. 9 inch dia. by 44.75 inches long. Double wrapped.
3. Nelson Muffler Corporation mufflers P/N T13928. 9.12 x 14.12 oval by 43 inches long. Double wrapped.

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7. AUTHOR(s)  Robert N. Baker		6. PERFORMING ORG. REPORT NUMBER
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)  Retrofit kits were developed to reduce acoustical noise at the operator's ear of the U.S. Army 10,000 Lb. Rough Terrain Forklift Truck. Modifications included replacement of exhaust mufflers, engine fan, hydraulic lines and a hydraulic pump. Engine, transmission and some hydraulic components were isolation mounted. Treatment also included use of sound barrier and absorption materials, panel damping and numerous special enclosure parts.		